(2)

N-1826

NCEL

Technical Note

March 1991

By J. Kunsemiller

Sponsored By Naval Facilities
Engineering Command

DEVELOPMENT OF A SEAWATER HYDRAULIC ROCK DRILL

ABSTRACT Under the Underwater Construction System program sponsored by the Naval Facilities Engineering Command, the Naval Civil Engineering Laboratory developed a seawater hydraulic tool system called the Multi-Function Tool System (MFTS). An objective of the MFTS development was to provide Underwater Construction Team divers with tools that meet their operational needs and were optimized for both the diver and the environment. The MFTS power source, bandsaw, rotary disk tool, and rotary impact tool completed the development cycle and have been released to the UCT divers for general use. A seawater hydraulic rock drill has not been issued because its performance is not predictable and its operation is not reliable. Extensive testing has isolated the problem to the impact mechanism and the timing of the cycle. Leakage of the supply poppet and the drive plunger as well as pressure pulsations from the rapid closing of the supply poppet have an undetermined negative affect on cycle operation. Cycle performance is not repeatable from test to test suggesting a transient or threshold condition that the drill is not always able to overcome. Additional development of the water lubricated impact mechanism is needed to investigate and correct these problems.

NAVAL CIVIL ENGINEERING LABORATORY PORT HUENEME CALIFORNIA 93043-5003

[redacted]

Approved for public release, distribution unlimited.

METRIC CONVERSION FACTORS

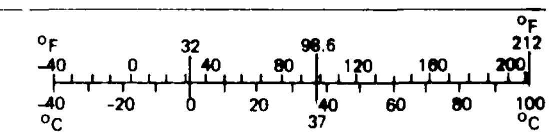
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



REPORT DOCUMENTATION PAGE	Form Approved OMB No. 0704-018

Public reporting burden for this collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection information, including suggestions for reducing this burden, to Washington Headquarters Services, Directorate for Information and Reports, 1215 Jefferson Davis Highway, Suite 1204, Arlington, VA 22202-4302, and to the Office of Management and Budget, Paperwork Reduction Project (0704-0188), Washington, DC 20503.

1. AGENCY USE ONLY *(Leave blank)*

2. REPORT DATE

March 1991

3. REPORT TYPE AND DATES COVERED

Final: Sep 1985 through Sep 1990

4. TITLE AND SUBTITLE

DEVELOPMENT OF A SEAWATER HYDRAULIC ROCK DRILL

5. FUNDING NUMBERS

PR - Y1606-001-A603
WU - DN387346

6. AUTHOR(S)

J. Kunsemiller

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESSE(S)

Naval Civil Engineering Laboratory
Port Hueneme, CA 93043-5003

8. PERFORMING ORGANIZATION REPORT NUMBER

TN - 1826

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESSE(S)

Naval Facilities Engineering Command
Alexandria, Virginia 22332

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

[redacted]

Approved for public release, distribution unlimited

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

Under the Underwater Construction System program sponsored by the Naval Facilities Engineering Command, the Naval Civil Engineering Laboratory developed a seawater hydraulic tool system called the Multi-Function Tool System (MFTS). An objective of the MFTS development was to provide Underwater Construction Team divers with tools that meet their operational needs and were optimized for both the diver and the environment. The MFTS power source, bandsaw, rotary disk tool, and rotary impact tool completed the development cycle and have been released to the UCT divers for general use. A seawater hydraulic rock drill has not been issued because its performance is not predictable and its operation is not reliable. Extensive testing has isolated the problem to the impact mechanism and the timing of the cycle. Leakage of the supply poppet and the drive plunger as well as pressure pulsations from the rapid closing of the supply poppet have an undetermined negative affect on cycle operation. Cycle performance is not repeatable from test to test suggesting a transient or threshold condition that the drill is not always able to overcome. Additional development of the water lubricated impact mechanism is needed to investigate and correct these problems.

14. SUBJECT TERMS

Seawater, rock drill, tools, diver equipment, hydraulic equipment

15. NUMBER OF PAGES

97

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	UL

CONTENTS

Page

APPENDIXES

INTRODUCTION

The Naval Civil Engineering Laboratory (NCEL) has developed tools, techniques, and equipment for improving the Naval construction divers' ability to work effectively underwater. The majority of Naval underwater construction is conducted by Underwater Construction Team (UCT) divers whose mission is to construct, inspect, maintain, and repair fixed Navy underwater facilities.

Under the Underwater Construction Systems (UCS) program sponsored by the Naval Facilities Engineering Command, NCEL developed a seawater hydraulic tool system called the Multi-Function Tool System (MFTS). An objective of the MFTS development was to provide UCT divers with tools that meet their operational needs and were optimized for both the diver and the environment. The MFTS power source, bandsaw, rotary disk tool, and rotary impact tool completed the development cycle and have been released to the UCT divers for general use. However, the most useful tool to the UCTs, the seawater rock drill, has not been released because of technical problems with the impact mechanism. The rock drill performance is not predictable and its operation is not reliable.

The objectives of this program were to determine the causes of the failures in the impact mechanism and to modify the rock drill so that it would meet the design requirements.

The scope of this investigation was limited to development of the double poppet-kicker port impact mechanism. No other impact mechanisms or drill designs were considered in this investigation.

BACKGROUND

Commercially available oil hydraulic tool systems have extended the capability of the UCT diver to do useful underwater work, however, there are disadvantages to their use (Ref 1). Oil leaks from the system can cause environmental contamination, pose a fire hazard, or threaten personal safety. Seawater leaks into the system can destroy system components, resulting in excessive maintenance and down time. In addition, the unwieldy dual transmission hoses burden the diver, particularly in a current or surge.

Since 1976, NCEL has been developing a hydraulic tool system that uses seawater instead of oil as the power transmission fluid (Ref 1, 2, and 3). The open-loop seawater hydraulic system provides the diver with easy to handle, single hose tools that are compatible with their environment. The system has all the benefits of oil hydraulic systems, and yet it does not present a health or fire hazard.

While seawater is an attractive alternative to oil from an application point of view, it provides many challenges in mechanical design. Aside from promoting corrosion in metals, the low viscosity of seawater offers minimal lubrication and high leakage rates compared to oils. These factors complicate the design by limiting material selection to those satisfying corrosion and lubrication criteria. In addition, close tolerance machining is necessary to

maintain reasonable operating efficiencies. These design considerations were especially evident in the rock drill.

This report begins with the design requirements for the seawater rock drill. This is followed by brief descriptions of the rock drill in the two configurations tested. Test results are then presented chronologically from each of three development test phases. Finally, a summarized list of findings and recommendations for further development are presented.

REQUIREMENTS

The design requirements for the seawater rock drill are given in Table 1. Each requirement represents a minimum threshold value of acceptable performance. The "as-built" characteristics are given for comparison. Several requirements were not evaluated on the rock drill due to development problems. The rock drill exceeded the weight requirement by 9 pounds. However, this weight increase does not appear to detract from the operability of the drill and may, in fact, increase its effectiveness. The 49-pound tool can drill into rocks of differing compressive strengths faster than a Stanley HD-20 oil hydraulic rock drill. The seawater hydraulic rock drill uses commercial twist and cross bits ranging in diameter from 3/4 inch to 2 inches.

COMPONENT DESCRIPTION

There were two major configurations of the rock drill used during this development. Both configurations use a poppet and kicker port hydro-mechanical linear impact mechanism. Variations to the following descriptions are discussed in the test and evaluation section as they occurred. An assembly drawing of the latest rock drill configuration is provided in Appendix A.

The first configuration, known as the Advanced Development Model (ADM), is shown in Figure 1. The ADM rock drill uses a double poppet-kicker port hydro-mechanical linear impact mechanism, Figure 2. A functional description of the double poppet-kicker port cycle is as follows. During tool operation, water at supply pressure enters the drill through the trigger and is directed into the drive chamber through the open supply poppet. This closes the exhaust poppet and drives the plunger and piston down into the drill steel, creating a percussive impact at the rock surface. Parallel to this drive stroke, pressurized water is directed through an orifice to the 3 hp seawater motor to rotate the drill steel. At the end of the drive cycle, the kicker port is pressurized, closing the supply poppet and allowing the exhaust poppet to open. This allows the the drive chamber to bleed down through the exhaust poppet and permits the plunger and piston to return to top dead center in preparation for the next cycle. This sequence is repeated approximately 2,750 times per minute. Exhaust from the linear impact system is directed out the side of the tool, while motor exhaust is ported out the top motor ports.

The ADM rock drill also included a built accumulator to enhance system performance. This accumulator was gas charged.

The second rock drill configuration, known as the Pre-Production Prototype (3P), is shown in Figure 3. The 3P rock drill uses a simplified single poppet-kicker port cycle, Figure

4. In the single poppet-kicker port cycle, the exhaust poppet is replaced with an orifice. A functional description of the single poppet kicker port cycle is as follows. During tool operation, water at supply pressure enters the drill through the trigger and is directed into the drive chamber through the open supply poppet. This drives the plunger and piston down into the drill steel, creating a percussive impact at the rock surface. At the end of the drive cycle, the kicker port is pressurized, closing the supply poppet. This allows the the drive chamber to bleed down through the exhaust orifice and permits the plunger and piston to return to top dead center in preparation for the next cycle. Exhaust flow from the linear impact mechanism is directed to the 3 hp seawater motor to index the drill steel. Motor exhaust is vented out a motor exhaust port.

The 3P rock drill also included a rotary type reversing valve to change the direction of motor rotation and an adjustable orifice control on the motor flow to control rotation speed.

TEST AND EVALUATION

The laboratory test and evaluation of the seawater rock drill was conducted at NCEL at the Seawater Hydraulics Laboratory and at the Ocean Systems Test Facility by Eastport International and NCEL personnel. Tests were conducted in rock samples of various compressive strengths. Human factors tests and field tests were conducted with Navy divers under supervision of NCEL engineers. Testing was conducted in three phases. Phase I testing was performed on the ADM drill. Phase II testing was performed using both the ADM and 3P rock drills. Phase III testing was conducted using a modified ADM rock drill.

PHASE I TESTS

Factory Acceptance Evaluation

All attempts at operating the ADM rock drill as delivered were unsuccessful. Individual drill components conformed to the drawing specifications. The overall quality of the fabrication and construction was good. Further investigation into the problem was necessary.

Operability Tests

The tool was configured with the double poppet-kicker port cycle and parallel motor porting at the start of the test program.

Initial Operating Tests. Initial attempts to start the tool were unsuccessful due to excessive seal friction on the supply poppet and piston. These seal clearances were enlarged by abrasive removal of seal material until the supply poppet and piston moved freely. The drill then ran roughly and intermittently. Further improvements involved adjustments to the cycle timing. Since the supply and exhaust poppet assemblies were each provided with several different springs of various spring rates, a series of tests were performed to determine the best spring rate. Through trial and error testing, the best supply and exhaust spring sizes were selected.

Material galling between the drive plunger and sleeves, between the supply poppet and sleeve, and between the exhaust poppet and sleeve became apparent. Design material specification was SCF 19 stainless steel for the drive plunger, supply poppet, and exhaust poppet. This was substituted with Nitronic 50 stainless steel. The design material for the sleeves was Stellite 6. This was substituted with Nitronic 60. The recommended design material was changed to Nitronic 50 or 60 combinations because of advertised galling resistance, material availability, and cost savings.

The most persistent problem involved the drive plunger and sleeve assembly. The Nitronic 50 plunger would seize in the Nitronic 60 sleeves after 2 minutes of operation. This combination, under conditions of low lubrication, did not provide the necessary anti-galling properties. In an effort to improve this sliding interface, a hard surface coating on the plunger was employed. The new plunger, made of 440C steel, was coated with Nedox, a self-lubricating material that provides a surface hardness of 68-70 Rc. Although this combination did fare better than the Nitronic pair and eliminated seizing, the Nedox coating began delaminating after approximately 5 minutes of operation. In addition, the 440C base material pit corroded when stored wet and required daily maintenance to remove the corrosion products.

It was also observed that the plunger impact area on the lower sleeve had plastically deformed, resulting in a reduced diameter and a rough edge. This rough edge was partially responsible for the failure of the Nedox coating. Chamfering this edge allowed the drill to run adequately, though the Nedox coating continued to delaminate. To decrease long-term deformation in the contact area, a hardened 440C insert was installed in the sleeve. This insert greatly reduced the rate of deformation.

The operation of the supply and exhaust poppet assemblies were improved, using the Nitronic combination, by a fine lapping of the poppet to the sleeve. Though this process helped to alleviate the problem, continued polishing of galling marks was required to maintain free operation.

A consultant specializing in hydraulic measurements, was brought in to help trouble shoot the timing of the linear impact mechanism. Timed pressure traces of the rock drill during operation (Ref 4) indicated that the timing and blow energy were not at design levels. Further testing identified two causes for the reduced performance.

First, it was demonstrated that back pressure of the linear impact mechanism exhaust flow had a negative affect on cycle timing. When this exhaust was ported through the drill steel for bit flushing, the back pressure caused rough and intermittent tool operation. When the exhaust was unrestricted to ambient, the drill ran smooth and steady. This arrangement prevented flushing through the drill steel.

Second, the built in accumulator intended to smooth out supply line pulsations did not function due to a variety of problems. When an external accumulator was attached to the drill, it was found that, with all other variables optimized, the accumulator made no measurable difference in drill performance.

The indexing rate of rotation of the drill steel affects the drilling rate in different compressive strength rock. In the seawater rock drill, with its fixed gear ratio, this rotation is controlled by the speed of the motor. An orifice placed in the motor supply line limits the motor speed by restricting flow. Several different orifice sizes were tested to obtain a motor speed for optimum drilling rate. A good average rate for the various diameter bits and compressive strength rocks tested was obtained using a 0.078-inch diameter orifice.

Performance Tests. A series of performance tests were conducted to determine drill rates with different bits in different compressive strength rock. The three types of rocks used in these tests were a soft igneous rock, a medium granite rock, and a hard basalt rock. The drill bits ranged from a 3/4-inch diameter twist bit to a 2-inch diameter cross bit. All tests were conducted in air.

The seawater hydraulic rock drill design requirement was to match the drilling rate of the Stanley HD-20 oil hydraulic rock drill currently used by the UCT's. For comparative purposes, holes were drilled in the soft igneous and medium granite rocks using the HD-20 with 3/4-inch and 1-1/2-inch diameter twist bits. (The HD-20 will not accept cross bit drill steels.) The seawater rock drill was operated using a 3/4-inch diameter twist bit and a 1-1/2-inch diameter cross bit. (The 1-1/2-inch diameter twist bit could not be used at this time with the seawater rock drill due to a difference in chuck size.) The respective drill rates are given in Table 2 and shown graphically in Figure 5.

The seawater hydraulic rock drill exceeded the rates established by the HD-20. It should be noted that the seawater drill, though operating satisfactorily, was not operating to design levels. Better drill rates were anticipated from an optimized rock drill.

The rock drill was also tested at different operating pressures and flow rates, ranging from 500 psi and 5 gpm to a maximum of 1,500 psi and 12 gpm. Although the drill did run at reduced pressures and flows, the severe reduction in drilling rates dictates that the drill be operated at the maximum flow of 12 gpm at a pressure of 1,500 psi.

Noise Level. Sound level measurements were conducted in open water with a hydrophone placed at distances of 1 and 2 feet from the tool. These are the distances of a buddy diver assisting to help start the bit and of the operating diver, respectively. The data were collected on tape and later analyzed to provide equivalent sound pressure levels in air (Ref 5). At a distance of 2 feet, the tool produced an adjusted sound pressure level of 80.4 dB referenced to 20 μPa. At a distance of 1 foot, the tool produced an adjusted sound pressure level of 96.0 dB referenced to 20 μPa. At this close proximity to the tool, the assisting diver would be exposed to more than the permissible exposure level (PEL) of 84 dB for 8 hours in any 24-hour period. The permissible exposure time for the assisting diver who is within 1 foot or less of the tool is calculated to be 1 hour in any 24 hour period. The noise level for the operating diver is within the PEL and is not a hazard.

Human Factors Evaluation. The rock drill was evaluated by three NCEL divers under actual operating conditions for handling, ease of use, control, safety, and other human factors considerations. Interview forms recording diver comments are provided in Appendix B, and are summarized below.

a. The balance and the weight of the tool were good. The drill seemed heavy enough to operate in the surf zone, and could even be weighted another 5 to 8 pounds for increased performance.

b. The palm trigger received high ratings for comfort and the lack of effort required when operating for extended periods.

c. The motor seemed underpowered when used with the larger bits, particularly in deeper holes.

d. The tool noise level was considered low to medium when compared to other hydraulic rock drills. The high frequency, low amplitude noise did not create any discomfort when the diver stood above the tool, but was slightly irritating when the diver was at the same level as the tool.

e. The supply hose did not pose any real hazard or discomfort to the diver, but a 90-degree elbow may make it easier to move about and to operate from either side.

Field Test at Anacapa Island

Field testing of the ADM rock drill was conducted at Anacapa Island by UCT-2 personnel. As a precaution against silt intrusion into the piston chamber which might jam the drill, the drill was wrapped in filter cloth material. At this point, the affects of silt intrusion into the piston chamber had not been tested. During this field test, the divers had difficulty handling the tool on the seafloor while carrying it across rocks from site to site. Several times, the trigger was actuated unexpectedly when the drill was lifted. Based on this occurrence, a requirement was added to provide an auxiliary lifting handle and/or a safety trigger.

Reliability Analysis

Endurance testing of the seawater hydraulic rock drill was conducted in air, not in water. This was to provide an adequate indication of the rock drill operational capabilities. However, because testing was conducted with fresh water, material corrosion proolems were masked. Material corrosion and unreliable performance later led to a second phase of rock drill development.

The endurance test was conducted in air using fresh water as the fluid medium. Drilling was conducted in various rock samples. The TEMP requires an 80 percent reliability. A 70 percent confidence level was agreed to by the sponsor as a reasonable test level. Forty eight hours of testing were required (Ref 6) at a 50 percent duty cycle, to verify the 40 hour mean time between failure (MTBF). Since there is no wear when the drill is off, the off portion of the cycle was shortened. This did not alter the critical running time or number of on/off cycles. The rock drill testing followed an accelerated duty cycle of 5 to 7 minutes on, 2 to 3 minutes off, with the off time being that required to allow the power source reservoir to refill.

During these tests, two components failed. Table 3 shows approximately when these failures occurred, the cause of failure, the time to repair the item, and subsequent corrective redesign action. The first came approximately 18 hours into the test (actual running time) and become apparent when the linear plunger/piston cycle ceased to operate. Inspection revealed that the drive plunger had fractured at the top of the reduced diameter of the annular cutout because of a stress raiser created during machining. The plunger was replaced and testing continued. At approximately 21 hours of actual running time, a failure in the motor stopped drill rotation. A check of the motor indicated that the failure was due to several vanes sticking

in their slots in the motor. It was determined that this sticking was the result of a machining error in the cam that created burrs on the rotor edges. These burrs eventually closed down the vane slot clearances and seized the vanes. After a cleanup of the vane slots, rotor, and cam, no further problems occurred.

It should be noted that many of the rock drill components had undergone more hours of actual running time prior to the endurance test. The drive plunger and sleeve assembly required frequent maintenance during these test. The maintenance consisted of polishing galling marks on the sleeves and polishing corrosion products off the plunger. The delamination of the Nedox coating continued but was not recorded as a failure since the drill still operated.

Phase I Modifications

The following modifications were incorporated into the 3P rock drill design. Two 3P rock drills were built.

1. Fabricate the drive plunger from 440C steel and coat with Nedox since this was the best material tested. The plunger should have a smooth surface finish with no machining grooves, which might form stress concentrations. The front plunger sleeve should have a hard insert to prevent sleeve deformation at the impact zone.

2. Install an auxiliary "D-shape" handle for lifting the tool. Install a trigger safety that prevents accidental engagement of the trigger.

3. Use Nedox coating on the supply and exhaust poppets to reduce the potential for galling. Increase the seal clearance on the supply poppet to reduce the seal friction between the poppet and the sleeve. Slightly increase the clearance between the supply poppet and the sleeve.

4. Increase the seal clearance with the piston to reduce seal friction between the piston and the sleeve.

5. Eliminate the accumulator system. Reconfigure the back flange and valve housing for weight and machining cost savings.

6. Reduce backpressure on the linear impact mechanism by porting cycle exhaust flow out the side of the tool in the area below the exhaust poppet.

7. Enlarge the lower piston ambient ports in the piston sleeve and housing to reduce possible dampening effect by the water cushion on the impact force between the piston and anvil.

8. Provide a means to filter out sand and sediment that might enter the piston chamber. The filter must permit free flow of water so as not to restrict piston movement.

9. Eliminate the relief valve in the rotary nose assembly.

10. Provide a porting plate or reversing valve to the motor and chuck adapters to allow convenient switches between twist bits and cross bits.

11. Provide an adjustable needle valve in the motor supply line to control motor speed to change the indexing rate of the drill.

PHASE II TESTS

Factory Acceptance Evaluation

Initial trials of the 3P rock drills demonstrated drill rates below the design requirement. The cause of this decreased performance was not readily apparent, but was partially due to the design changes incorporated in the 3P model. At the start, five deficiencies on the 3P drills were identified and corrected as described below:

1. The Nedox coating on the supply and exhaust poppets adversely altered poppet timing. This coating was applied, based on use on the drive plunger, to prevent galling. The coated poppets were replaced with uncoated poppets.

2. The hex chuck and shank adapters interfered with the anvil housing and damaged the anvil bore such that the anvil was not allowed free movement. To correct the design, a radius was added to the anvil housing to eliminate this contact. In addition, a stop was added to the chuck to prevent the shank adapter from contacting the anvil housing.

3. The hard insert was inadvertently omitted from the front plunger sleeve. The deficiency was corrected by reworking the sleeve.

4. The trigger safety did not completely prevent the trigger plunger from unseating when the palm trigger was depressed. As a result, the small amount of flow past the trigger plunger was sufficient to start rotation of the drill steel. A safety analysis (Appendix C) supported eliminating this feature in favor of additional training in safe handling of the tool.

5. The throttle valve, intended for adjustment of the drill indexing rate, had no noticeable affect on rotation speed. The reversing valve, added to change bit rotation, leaked water, stalling the motor. To correct these deficiencies, the throttle valve was eliminated and a porting block replaced the rotary reversing valve.

Operability Tests

An instrumented test stand, Figure 6, was constructed to aid in collecting data on rock drill performance. Relative values of impact energy were used to compare impact energy from test to test. The stand was configured to hold the tool upright and permit unrestricted vertical movement. As in actual operation, the weight of the tool provided the reaction force

to the impact of the drill steel on the stand. A force transducer mounted to the impact plate measured the percussive force and the conditioned signal was recorded on a strip chart recorder in an amplitude versus time format.

The force spikes directly correlated to the impact energy transmitted by the linear system in the tool. By comparing the strips from test to test, the affect of a change to the tool, as it pertains to impact force, could be determined. The recorder also displayed pressure traces taken at selected locations on the tool along the same time scale.
This provided analytical information on the timing sequence of the linear system in relation to the impact force. A sample trace is shown in Figure 7.

ADM Operating Characteristics. The objective of this test was to compare ADM to 3P performance to find out why the ADM rock drill was superior. Our approach was to compare drill performances using the same internal components. A Stanley HD-20 oil hydraulic rock drill was used for drill rate comparison. Although the ADM drill had previously surpassed the HD-20 performance level of 4 inches per minute during Phase I testing, the ADM drill rate was now less than 2 inches per minute. Instrumented tests showed little impact force being imparted to the drill steel by the linear mechanism and low drive chamber pressure (drive chamber pressure directly relates to impact force).

Further investigation with the ADM rock drill showed that the exhaust port pressure closely followed the drive chamber pressure (see Figure 8). This indicated that the exhaust poppet was not fully closing during the cycle. It was theorized that the combined affects of exhaust poppet leakage, drive plunger leakage, and parallel motor porting reduced the supply to the linear drive system. This would account for the low drive chamber pressure. To verify this, the motor supply was blocked so that the full supply flow of 13 gpm was directed to the linear system. As expected, the drive chamber pressure increased, as shown in Figure 9.

The data collected during the tests were inconsistent and often not repeatable. Several factors seemingly combined to vary the results. Most particularly was the degradation of the Nedox coating on the drive plunger, Figure 10. The mechanical removal of the coating was immediate and continuous. As a result, leakage rates and cycle timing were unpredictable.

Three significant results were recorded. These were:

1. The ADM no longer meet the design requirement for drill rate. This dramatic decrease in performance from earlier Phase I results was attributed to increased leakage rates. The drive plunger coating degradation was the primary cause of these increased leakage rates. Variations in tool performance, even between identical configurations in consecutive tests, were indicative of the complexity of the system.

2. The exhaust poppet was not operating as designed because it did not completely close prior to the drive stroke.

3. The supply flow was insufficient, with the motor in parallel to the linear impact system, to fully energize the system.

Design Review. Consultants were brought in to review the tool design and identify critical parameters that affect the cycle operation. It was not a redesign effort, but was limited to determining potential enhancements to the existing design.

Both consultants concluded that the double poppet-kicker port cycle was a poor choice for this tool and they doubted that it could be made to work efficiently. The cycle has too many critical parameters to accurately control with only hydraulic forces available, particularly at cycle rates up to 3,000 per minute. Several specific problem areas were identified.

First, the poppet valves could not be properly balanced. The spring rate needed to balance the poppets during the cycle was not constant, particularly for the exhaust poppet. A weak spring was needed to allow the poppet to open, while a strong spring was required to hold it closed. With no mechanical control (only hydraulic and dynamic influences), poppet operation varied from cycle to cycle.

Second, high leakage and motor flow rates combined to reduce flow to the linear drive system. This resulted in low drive chamber pressure and reduced impact energy to the drill steel. The design clearance between the drive plunger and sleeves was necessary to accommodate thickness variations of the Nedox coating. Once the coating began to fail, this clearance became excessive and the leakage rate increased.

Third, the Nitronic 50 and 60 materials used in the supply and exhaust poppet assemblies and the drive plunger sleeves was not suitable to withstand the cyclic impact loads. The results were manifested in the supply poppet and drive plunger assemblies, where impact surfaces repeatedly deformed.

Fourth, the location and size of the timing and flow ports in the drive plunger sleeves, particularly the kicker port gallery, was questioned. Improper location and insufficient gallery length would limit the contact time between ports, truncating the cycle and detuning the circuit. The overall effect, depending on the severity of the condition, could range from a slightly less efficient tool to one that would not cycle at all. The original locations were chosen as starting points with the intention of optimizing them during testing. However, the transient nature of the tool performance made optimization extremely difficult at this point.

At the conclusion of the design review process, three recommendations to increase the performance levels of the tool to an acceptable level were offered. Copies of the Design Review Reports submitted are included in Appendix D. The recommendations were:

1. Port the motor in series with the linear impact system by using the exhaust flow to supply the motor. This would ensure that the linear system received sufficient flow to achieve the maximum power output from the drive cycle.

2. Use high strength, high hardness materials for the linear impact mechanism components. Using materials like Stellite and SCF19 steels should preclude the need for coatings and minimize the effects seen with coating degradation (additional leakage, variable friction, damage to components). These materials should also provide the strength and hardness needed to withstand the impact forces.

3. Maintain tighter clearances and tolerances as specified in the original ADM design to reduce leakage. The use of the superalloys listed above would assist in maintaining these clearances that are so critical to the successful operation of the tool.

ADM Serial Configuration Test. To test the first design review recommendation, the exhaust flow from the exhaust poppet was ported to the motor. Backpressure affects on the linear impact mechanism were minimized by using a short hose length and large internal

diameter fittings. This increased the drill rate to an average of 5 inches per minute using a 1-inch twist bit. This rate exceeded the HD-20 rate by 25 percent. It was observed that a masonry auger yielded a 20 percent higher drill rate than a standard twist drill bit with either the HD-20 or the ADM for the type of rock used in the tests. The masonry auger was thereafter used in all drill rate tests.

During these tests the drive plunger broke at the diametrical transition where it was impacting against the front sleeve. This fatigue failure was the result of repeated impacts against the front plunger sleeve. The front plunger sleeve insert continued to deform from impacts and required occasional rework to allow the plunger to move freely.

3P Parts Testing. The ADM was assembled using new poppet, plunger, and piston assemblies from a 3P drill. The ADM drill cycled well but provided very little impact energy to the drill steel. The worn and damaged ADM parts worked better. Parametric tests identified the supply poppet as the the component most responsible for the reduced impact energy. When the ADM supply poppet was replaced with a new one, and all other factors held constant, the performance decreased, as recorded by low force spikes.

Figure 11 shows the difference in the pressure traces between the old poppet and the new poppet. The new poppet has a higher than normal, very short duration pressure spike in the piston supply line at the bottom dead center point in the cycle. After some review, it was hypothesized that this pressure pulse, the result of the rapid closure of the new supply poppet, was squeezing the piston seals to brake the piston just prior to impact against the anvil.

A comparison of the new and the old supply poppet assemblies showed the physical difference to be a worn-in seat area on the old poppet face, Figure 12. This worn-in area was the result of repeated poppet closures on the poppet seat. To duplicate this difference, the new poppet was lapped to approximate the profile of the groove in the original poppet, Figure 13. The grooved poppet restored impact energy. Evidently, the groove provides a cushion to attenuate the water hammer that causes the piston seals to brake the piston. The full dynamics of the seat profile are not understood, however, the desired seat profile was easily duplicated. All other poppets were subsequently lapped and matched to a seat with acceptable results.

In addition, the piston seals in the 3P rock drills were honed until the piston could drop freely under its own weight through the two sets of seals (about 0.003 to 0.005 inch diametrical clearance). Although more leakage was expected, very little additional flow past the seals was noted during testing, and no adverse effect on the operation of the tool was noted.

3P Drill Tests. One pre-production prototype rock drill designated 3P-A was tested using the 3P parts proven in the ADM drill. The 3P-A rock drill operated well in a serial configuration. It achieved drill rates of up to 6 inches per minute with a 1-inch diameter twist bit, almost 50 percent better than the HD-20. However, variations in performance were noted between different combinations of internal parts. Pressure traces on the test stand were often inconsistent and not repeatable. Bit seizure due to insufficient motor torque occurred in deeper holes, particularly when using 2-inch diameter drills.

Submerged Testing. This test was performed to determine what decrease in drill performance could be expected when the drill is submerged. The test stand was placed in a seawater tank deep enough to cover the rock drill. The sealed force transducer provided

relative impact energy. The 3P-A rock drill cycled consistently and smoothly though the force spikes were 10 to 15 percent smaller than those produced with the tool running in air. This decrease in performance was the result of reduced flow out the piston chamber ports providing a water cushion at the bottom of the piston stroke.

Filter Plug Evaluation. This test was performed to evaluate sintered metal filter disks as a means to prevent fine silt and sediment from entering the drill cavities and creating additional wear on the piston and drive plunger assemblies. With 100 micron plugs installed in the piston chamber ports, very little impact force was produced. Inspection showed that two of the filter disks had shattered and others showed varying degrees of deformation. The flow rate out of the piston chamber required more area than was available, therefore, the disks were discarded. If necessary, some other type of filter element needs to be used to cover these ports.

Field Testing at Guantanamo Bay

UCT-1 trained with the 3P-A rock drill in Guantanamo Bay harbor with a NCEL representative on hand to provide technical assistance. The tool operated well for about 15 minutes before stopping. Attempts to restart the tool were unsuccessful. The trigger valve was stuck partially open apparently caused by intrusion of fine sediment. No attempt to disassemble the trigger was made on site. The tool was returned to NCEL for inspection. Prior to disassembly, the 3P-A rock drill was placed in the test tank and operated. The tool started immediately and ran well with excellent force spikes. No trigger malfunctions occurred during this test, and no residual effects of the failure at Guantanamo Bay were apparent. Even when silt and sediment were added to the tank water to replicate the field conditions, there were no identifiable problems or failures. No direct conclusions could be drawn from this failure since the cause of the failure could not be determined. Under these circumstances, no corrective action could be taken.

Reliability Analysis

Endurance Test. A test to determine if any component failures or significant change in performance would occur during a typical operation scenario was conducted on the 3P-A rock drill. The test consisted of running the tool submerged in seawater on an automated cycle of 7 minutes on, 3 minutes off. This simulated a reasonable evolution for a diver drilling multiple holes of average depth in the field. The test ran 5 consecutive days to simulate a field deployment cycle. The total tool on time during this 5-day period was 10 hours. At the end of each day, the tool was removed from the tank and operated for 3 minutes on fresh water to flush the system. At no time during the 5-day cycle was the tool disassembled.

Prior to the start of the test, performance tests on the instrumented test stand determined the best combination of linear impact mechanism components and a baseline performance level was recorded. The same parts were used throughout the 5-day program in order to determine any degradation in the coating (Nedox failure), material incompatibilities (galling and corrosion), or other failure mechanisms that would affect the performance or projected life of the tool.

Throughout the 5-day cycle, the tool ran smoothly, although the supply pressure, measured at the hose reel, dropped from 1,450 psi to 1,350 psi. At the end of the 5-day cycle, the 3P-A tool was tested on the instrumented stand and it was found that the average height of the force spike had decreased by 25 percent from the pre-test level, and the in-air drill rate decreased by 35 percent.

Inspection revealed that the exhaust poppet had seized in the fully open position in its sleeve. This contributed to the drop in supply pressure (more leakage) and decreased drill performance. Further tests with a working exhaust poppet failed to completely restore the force spikes and drill rate to their pre-test levels because the Nedox coating on the drive plunger had degraded substantially. In addition, the insert on the front plunger sleeve showed significant deformation (see Figure 14).

PHASE III TESTS

A third phase of rock drill development was conducted to resolve continuing exhaust poppet failures and degradation of the Nedox coating on the drive plunger. This development was conducted on the ADM rock drill.

Exhaust Poppet Replacement

The failure of the exhaust poppet to close, as encountered during the endurance test, did not affect cycle timing. The apparent effect of the open exhaust poppet was a reduced drive chamber pressure. This resulted in a small reduction in impact energy. It was theorized that the linear impact mechanism could be simplified by replacing the exhaust poppet with an orifice. Tests were conducted with an adjustable orifice to determine an optimum orifice size that would produce a minimal decrease in drill performance. Parametric tests determined the optimum opening and a fixed insert having this configuration was made to replace the poppet assembly (see Figure 15). This simplified cycle is referred to as a single poppet-kicker port cycle.

Material Investigation

Corrosion free materials are critical in the drill assembly because the design clearances can not accommodate corrosion build up. To determine the corrosion effects from wet storage, the 3P-A rock drill was operated in seawater and then stored in air for 30 hours without disassembly or freshwater rinse. At the end of 30 hours, the rock drill was retested. The drill started and ran smoothly with no degradation in performance. The test was repeated for an additional 24 hours. Again, the drill started and ran smoothly with no degradation in performance. Based on this test, the rock drill demonstrated a tolerance to short term wet storage without adverse effects. However, long term wet storage, on the order of weeks, has proven fatal to drill operation.

Severe corrosion of the hardened 440C steel piston, anvil, and drive plunger during long term wet storage prompted an investigation of improved materials for these components. MP35N multi-phase stainless steel, a corrosion free material in seawater, was identified for

the piston, anvil, and drive plunger. To complement a MP35N drive plunger, Stellite 6B was used as the material for the plunger sleeves. This combination was to provide anti-galling and impact resistance under the conditions of tight clearances and seawater lubrication. Specifications for these two superalloys are given in Appendix E.

Prior to testing the superalloy parts, the ADM was checked to ensure that it was operating correctly. Then, the drive plunger and sleeve pair were replaced with the MP35N plunger and Stellite 6B sleeve pair. When operated, the MP35N drive plunger seized in the sleeves on the first down stroke. Inspection revealed galling in the sleeve bore. Damage to both the plunger and the sleeves was significant.

Tests of the MP35N piston and anvil showed the anvil deformed from impacting on the drill steel. After 14 hours of operation, the anvil had deformed to a depth of 0.016 inches. The hardness of the anvil was measured at 47 - 48 on the Rockwell C scale. Further heat treatment raised this to 50 Rockwell C, near the material limit for hardness. Additional testing with this anvil showed an additional 0.003 inch depression in less than 2 hours of operation.

From these results it was concluded that MP35N is not suitable for the anvil and may not be suitable for the drive plunger or piston because of its low hardness characteristic. The 440C steel components were replaced in the rock drill.

Drive Plunger Redesign

The galling experienced with the MP35N plunger and sleeve set drew attention to sleeve pair alignment and the plunger passage through the sleeve bore. This galling, and certainly the earlier failures of Nedox coated plungers, strongly suggested that the necessary alignment was not present. Maintaining sleeve alignment throughout the cycle is critical to the successful operation of the drive plunger system.

The problem is described as follows and illustrated in Figure 16. The drive plunger is formed by two diameters; a small diameter with a annular cut out midway on its length to facilitate fluid passage between ports, and a larger diameter cap to stop the plunger at the bottom of the stroke. The rear plunger sleeve is bored to the larger diameter and the front plunger sleeve is bored to the small diameter. The sleeves float on O-rings within the housing bore and they are clamped together, crushing the O-ring between them. Under these conditions, the sleeves could move in relation to one another, creating a misalignment of the bore. At the plunger position shown in Figure 16, the contact length between the plunger and the rear sleeve is minimal, which further contributes to misalignment.

The first attempt to control sleeve alignment, using a plunger of a single diameter, was unsuccessful. The new plunger design, Figure 17, having a single diameter along its length and thin bridge sections at the porting cutout, did not maintain contact with the rear sleeve throughout the stroke. When tested, both the 440C and the MP35N plunger and sleeve pairs failed within a few cycles. The alternative solution was to control alignment with a single sleeve.

A single sleeve, Figure 18, having the original bore profile of two diameters ensured the best possible alignment. This single sleeve was made from MP35N and tested with 440C/Nedox coated drive plungers. A marked improvement in the integrity of the Nedox coating on the plunger was observed even after 75 minutes of operation. An occasional chip to the coating was caused by the brittle Nedox coating reacting to the impact.

The single sleeve was successful at eliminating misalignment. Drill performance however, continued to be irregular, even between tests with identical components. Testing was discontinued at this point.

FINDINGS

The extensive tests conducted on the ADM and 3P model rock drills provided the following findings.

1. The rock drill in-air weight exceeds the design requirement by 9 pounds. This increased weight did not decrease drill performance or operability but it may have enhanced them. From an operability standpoint, the weight, balance, and noise level of the drill were acceptable.

2. The built-in accumulator system did not work nor did it provide any improvement to drill performance. An external accumulator did not produce measurable improvement either.

3. Porting the linear impact mechanism exhaust flow through the drill steel for hole flushing created an unacceptable backpressure on the linear impact system. The restricted flow adversely changed the cycle timing. Best drill performance was achieved when system backpressure was minimized.

4. Serial routing of the linear impact mechanism exhaust flow to the seawater motor improved drill performance because flow to the linear impact mechanism was increased. Parallel routing of flow to the motor and the linear impact mechanism required a flow rate greater than what could be supplied by the power supply. Optimum drill performance was achieved at an operating pressure of 1,500 psi and 10 gpm flow rate in the serial flow configuration.

5. The impact of the drive plunger on the top of the front plunger sleeve plastically deformed the sleeve. This diminished the diametrical clearance between the plunger and sleeve and resulted in seizure of the plunger in the sleeve. A hard metal insert in the front sleeve reduced the deformation and eliminated this problem.

6. The double poppet-kicker port cycle was simplified to a single poppet-kicker port cycle by replacing the exhaust poppet with a fixed orifice. Performance tests with a fixed orifice exhaust provided satisfactory results.

7. The seawater rock drill occasionally demonstrated drilling performance that exceeded the minimum design requirements. However, this performance was not repeatable from test to test. Increased leakage caused by larger clearances, primarily between the drive plunger and sleeves, accounted for some reduction in drill performance. However, no cause was determined for the variation in performance.

8. The single sleeve provided the necessary alignment for free passage of the drive plunger. The mechanical removal of the Nedox coating from the drive plunger, caused by earlier sleeve misalignment, was reduced to an occasional chip with the single sleeve.

9. The MP35N multi-phase stainless steel was evaluated for the anvil, piston, drive plunger and single sleeve as a replacement for 440C steel, which exhibited unacceptable corrosion. Tests with the anvil demonstrated that MP35N is too soft and critical dimensions could not be maintained under repeated impacting.

CONCLUSIONS

The seawater hydraulic rock drill does not meet design requirements because performance of the linear impact mechanism is not predictable. The feasibility of the poppet-kicker port cycle has not been proven. Though the single poppet-kicker port cycle is simplified from the earlier double poppet-kicker port cycle, the operation remains complex. What has been demonstrated is that the poppet-kicker port design is overly sensitive to variations in clearances. Therefore, a poppet-kicker port cycle design may not be suitable for this application.

Several factors were demonstrated to affect cycle timing and drill performance, the most significant of these was clearances and the relation to leakage. Large clearances mean more leakage and less usable energy. This was especially evident as the clearance increased between the drive plunger and the plunger sleeves as the Nedox coating was failing. For this mechanism to be successful, leakage must be kept to a controlled minimum in order to maintain the cycle timing.

No explanation was found to explain why drill performance is not repeatable from one test to the next. The suspected cause was thought to be the continual removal of the Nedox coating from the drive plunger leading to increases in the leakage rate. This would certainly explain a general decrease in tool performance but it would not account for the random good and bad performance exhibited. The successful single sleeve test indicated that the variation is caused by something other than the drive plunger clearance. It is likely that a transient or threshold condition may be preventing the drill from achieving expected performance.

The harsh environment seen by the internal drill parts requires materials that can withstand severe sliding and impact loads. The close clearances required by the drive plunger and the supply poppet to achieve proper cycle timing also promote material galling at the sliding interface. Materials like Nitronic 50, Nitronic 60, and MP35N, selected because of their antigalling characteristics, were not suitable for repeated impact loads because they are too soft and they deform. Grade 440C steel has the requisite hardness but is subject to pit corrosion in the seawater environment. The by-products of corrosion reduced mating part clearances and frequently rendered the tool inoperable.

Application of coatings such as Nedox, titanium nitride, and titanium carbide do little to protect the base material. Nedox coating seemed to offer the required lubricity for a sliding surface but behaved like a thick ceramic coating and was easily chipped away. It is apparent that the requirements for components in this rock drill design will require imaginative application of the latest material development to overcome heavily loaded sliding contact with little lubrication.

Finally, several design requirements have not been evaluated because of the development problems discussed. Most of these relate to reliability of the tool or the time required for maintenance. It seems appropriate, however, to raise a concern with the requirements for cold temperature storage and operation. It is unknown at this time what effects freezing water might have on the internal components as it might affect design clearances. Out of water operation at -1°C could lead to icing in the restrictive flow paths and prevent the tool from operating. At this time, only the storage of the motor at -20°C has been evaluated. Performance of the rock drill at freezing temperature is unknown at this time.

RECOMMENDATIONS

Continued development of a seawater hydraulic rock drill should begin with an investigation of alternate impact mechanism designs. A mechanism more suitable to seawater hydraulic application may have been overlooked. An option that should be considered is the use of a high water base fluid as the hydraulic fluid. This may allow a larger selection of materials since some lubrication is provided with this type of fluid.

If the single poppet-kicker port cycle continues to be the design of choice, development should continue independent of the rock drill. The objective should be to identify the conditions, such as alignment of coaxial parts, diametrical clearances, and linear position requirements for timing ports, that affect cycle performance. Incorporation into the rock drill should not be undertaken until a complete understanding of these conditions and their perturbations is obtained.

A model constructed of the linear drive system would be a useful tool to evaluate cycle characteristics prior to a laboratory demonstration. A significant cost savings might be realized by using a model to conduct the "what if tests." Then, final configuration could be validated to confirm the model.

REFERENCES

1. S.A. Black. "Seawater hydraulic systems for underwater equipment," paper presented at the Offshore Technology Conference, Houston, TX, May 1981 (OTC Paper No. 4084).

2. Naval Civil Engineering Laboratory. Technical Report R-895: Seawater hydraulics; development of an experimental vane motor for powering diver-held tools, by S.A. Black. Port Hueneme, CA, Jul 1982.

3. S.A. Black. "Development and evaluation of an experimental seawater hydraulic tool system for U.S. Navy divers," paper presented at the Offshore Technology Conference, Houston, TX, May 1984. (OTC Paper No. 4663).

4. Morris Engineering Company, Letter Report: Pressure pulsation of tests on rock drill at NCEL, by R.P. Morris, Fillmore, CA, 21 Jan 1986.

5. NEDU ltr 6420, Ser 404 of 29 October 1982, Subj: Diver tool noise conference summary.

6. Advanced Technology, Inc., Test Plan for MFTS, December 1984.

Table 1. Rock Drill Design Requirements

Item	Design Requirement	As-Built
Drill Holes in Coral and Rock	30 in. coral	
	12 in. rock	12 in. rock
	To 1 in. Dia.; Twist Bit with 3/4 in. Hex Shank Clockwise Rotation	met
	To 2 in. Dia.; Cross Bit with 7/8 in. Hex Shank Counter Clockwise Rotation	met
Drill Rate	Equivalent to HD-20	met
Weight in Air	40 lb	49 lb
Operating Depth	190 ft	unlimited
Operating Temperature	-1 to +40 °C	
Storage Temperature	-20 to +60 °C	
Reliability (R)	0.80	
MTBF	36 hr	
MTTR	1 hr	
Availability	0.80	
Maintenance:		
Daily	0.5 hr	
Project end	1 hr	
Annually	4 hr	

Table 2. Comparative Drill Rates

Bit Diameter	Drill Rate (in./min)			
	Soft Igneous		Medium Granite	
	HD-20	Seawater	HD-20	Seawater
3/4-in. twist	3-1/2	4	3-1/8	3-1/2
1-in. twist	2-1/2	-	2-1/8	-
1-1/2-in. twist	1-7/8	-	1-5/8	-
1-1/2-in. cross	-	2-1/4	-	2
2-in. cross	-	1-5/8	-	1-1/4

Table 3. Rock Drill Life Test

Running Time (hr)	Failed Item	Cause of Failure	Time to Repair	Redesign Action
18	Plunger	Fatigued and sheared	15 min	Finer surface finish
21	Motor	Vane seizure	1 hr	None

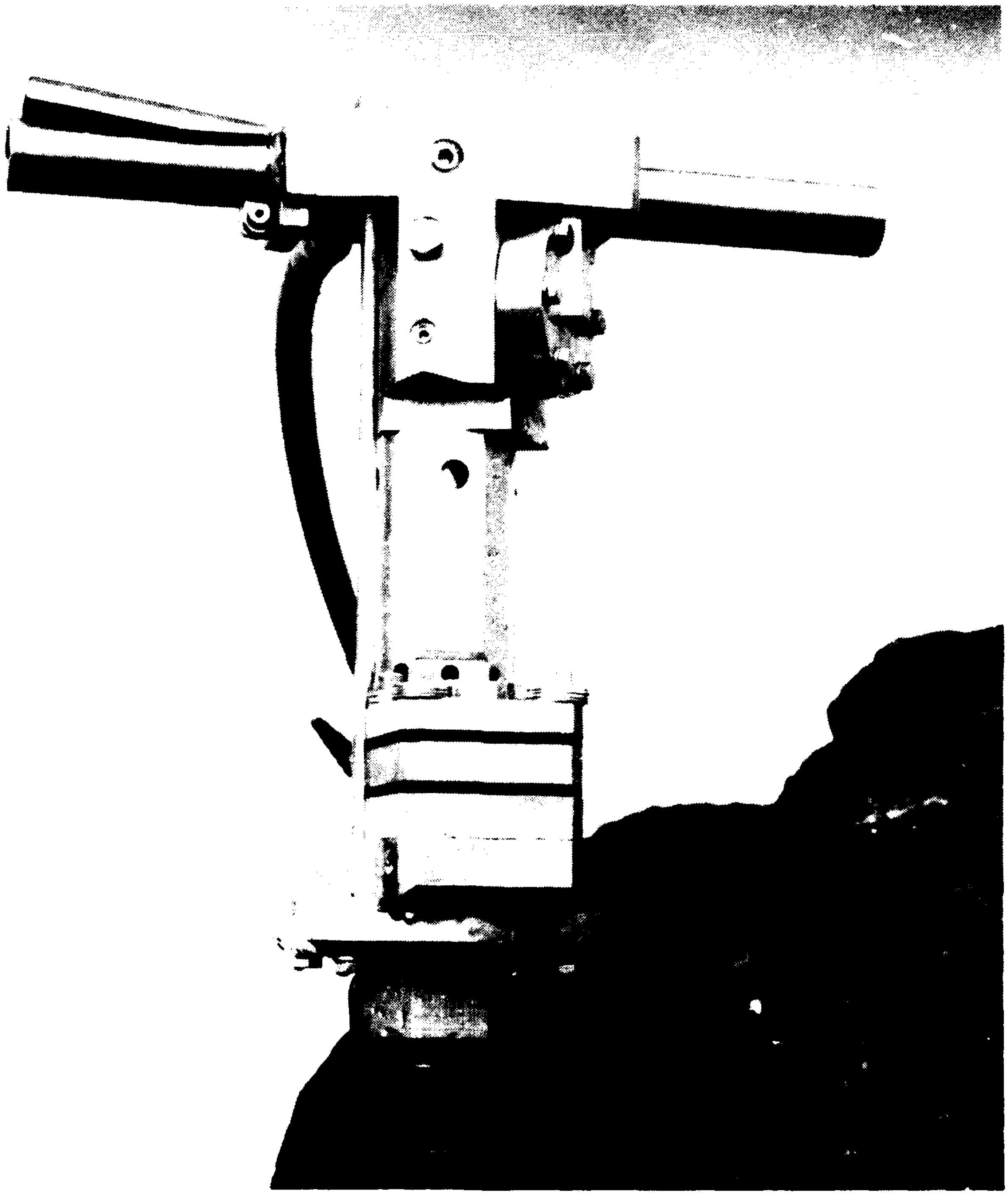
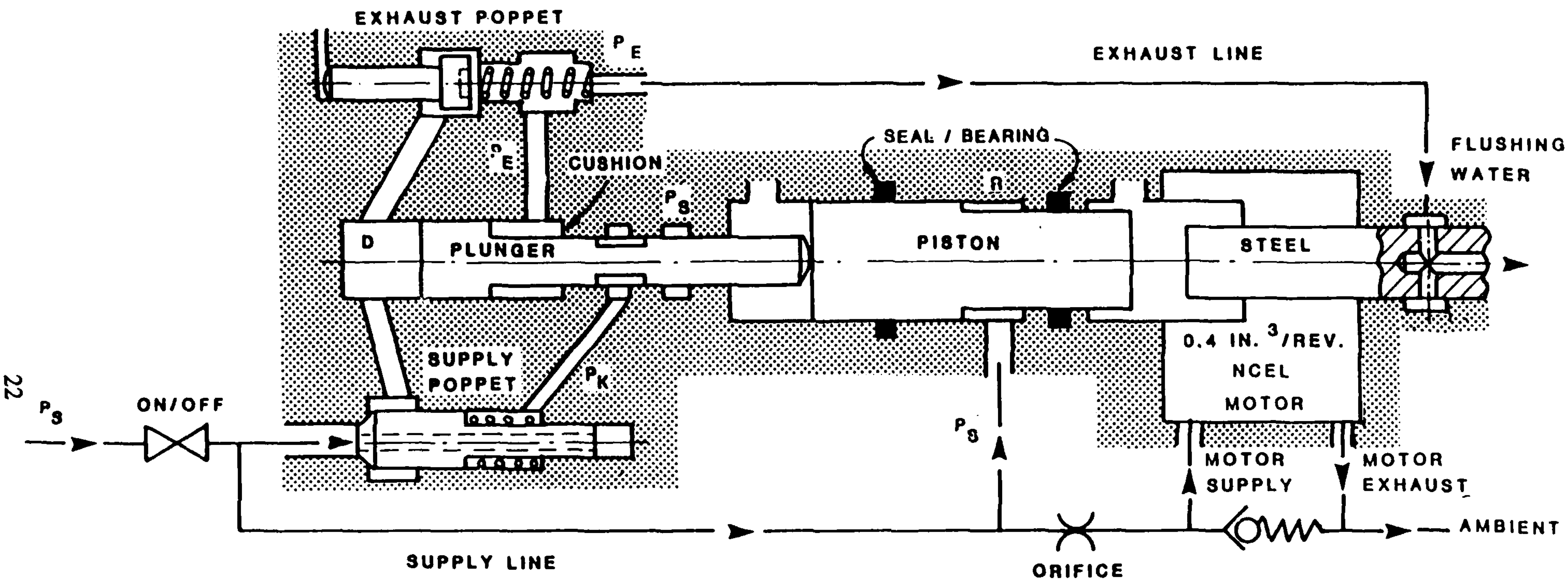


Figure 1. ADM rock drill.

Kicker Port Cycle and Hydraulic Circuit
Parallel Configuration



D = DRIVE END
R = RETURN END

P_S = SUPPLY PRESSURE
P_E = EXHAUST PRESSURE
P_K = "KICKER" PRESSURE

Figure 2. ADM rock drill with a double poppet-kicker port hydromechanical linear impact mechanism.

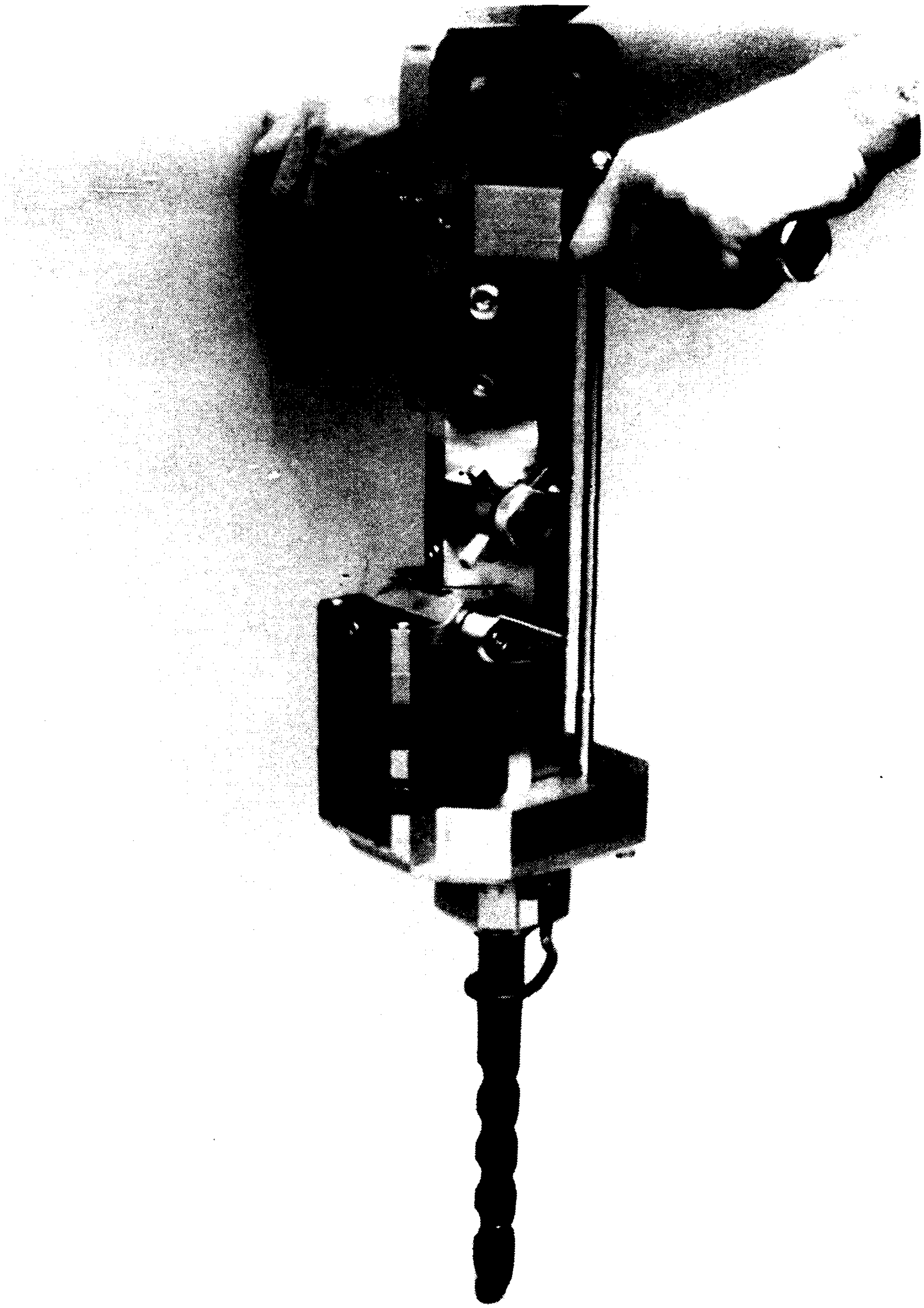
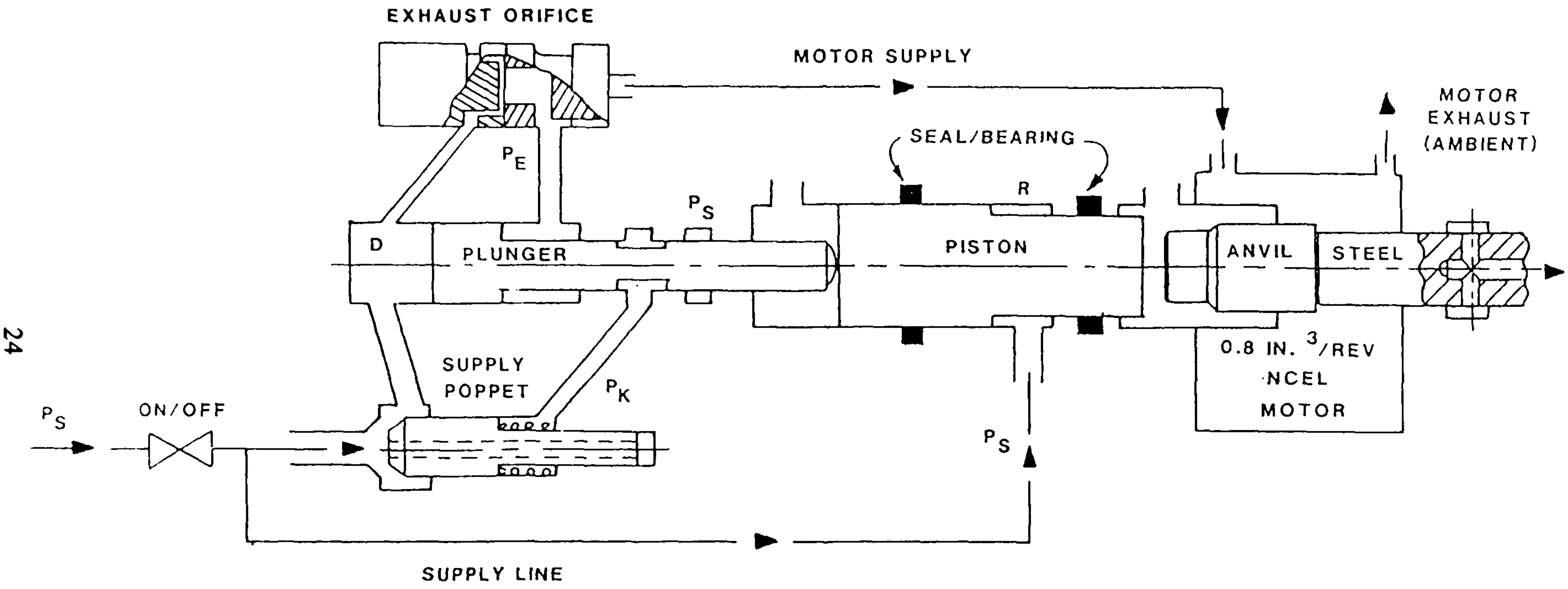


Figure 3. Pre-product prototype (3P) rock drill.

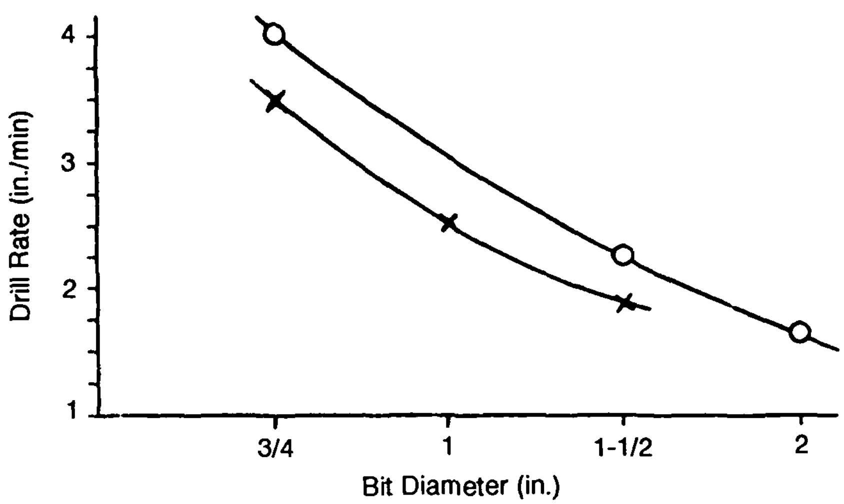
Kicker Port Cycle and Hydraulic Circuit
Serial Configuration



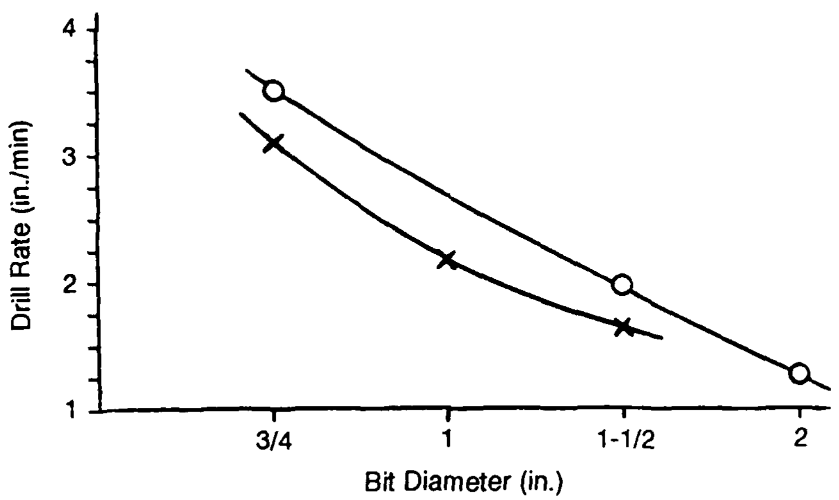
D DRIVE END | P_S = SUPPLY PRESSURE | P_K = "KICKER" PRESSURE

R RETURN END | P_E = EXHAUST PRESSURE

Figure 4. 3P rock drill with a simplified single poppet-kicker port cycle.



(a) Soft Igneous Rock



(b) Medium Granite

X = HD-20 oil hydraulic rock drill
O = NCEL seawater hydraulic rock drill

Figure 5. Comparative drill rates.

Figure 6. Instrumented test stand.

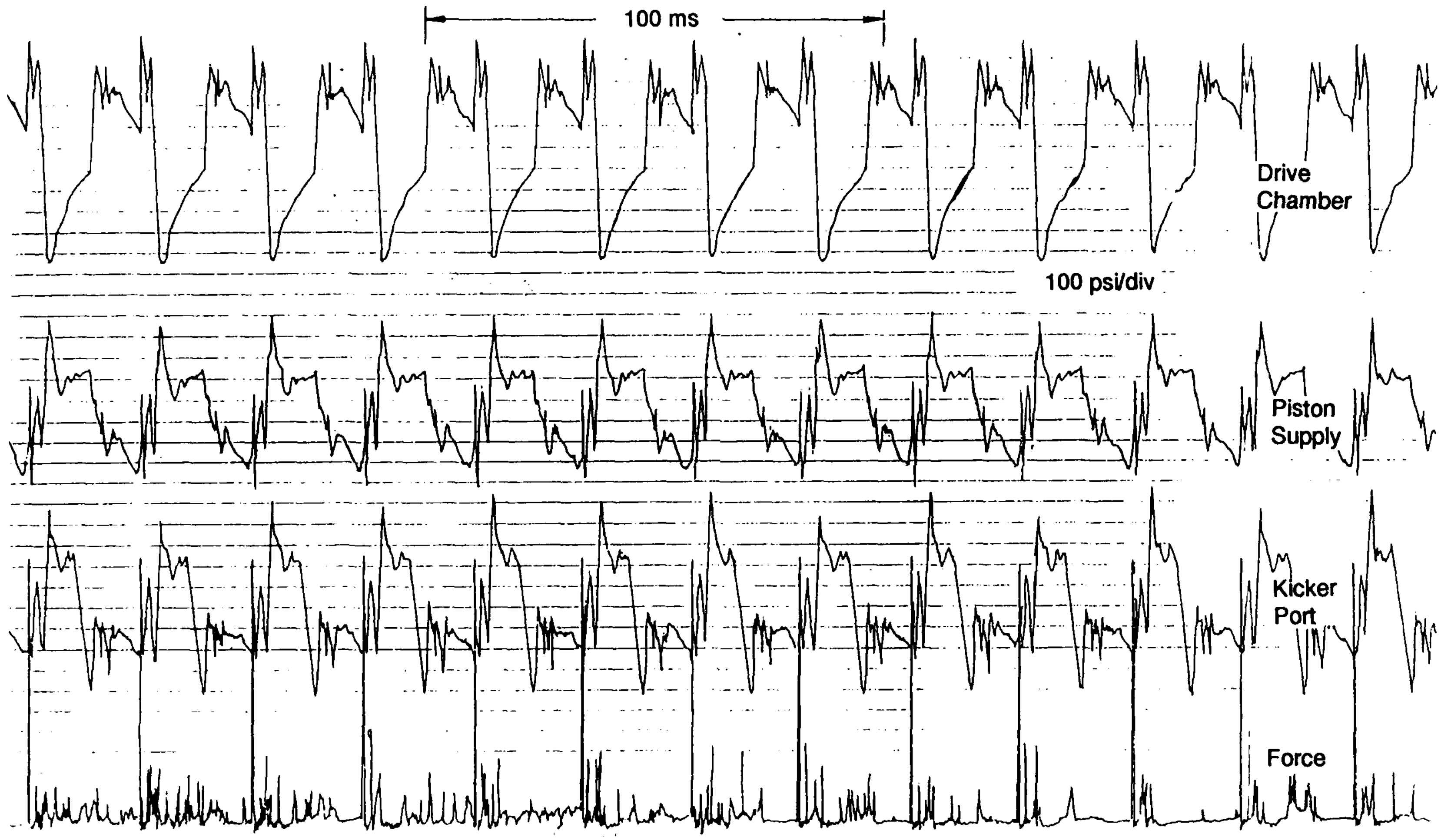


Figure 7. Sample of seawater hydraulic rock drill pressure traces.

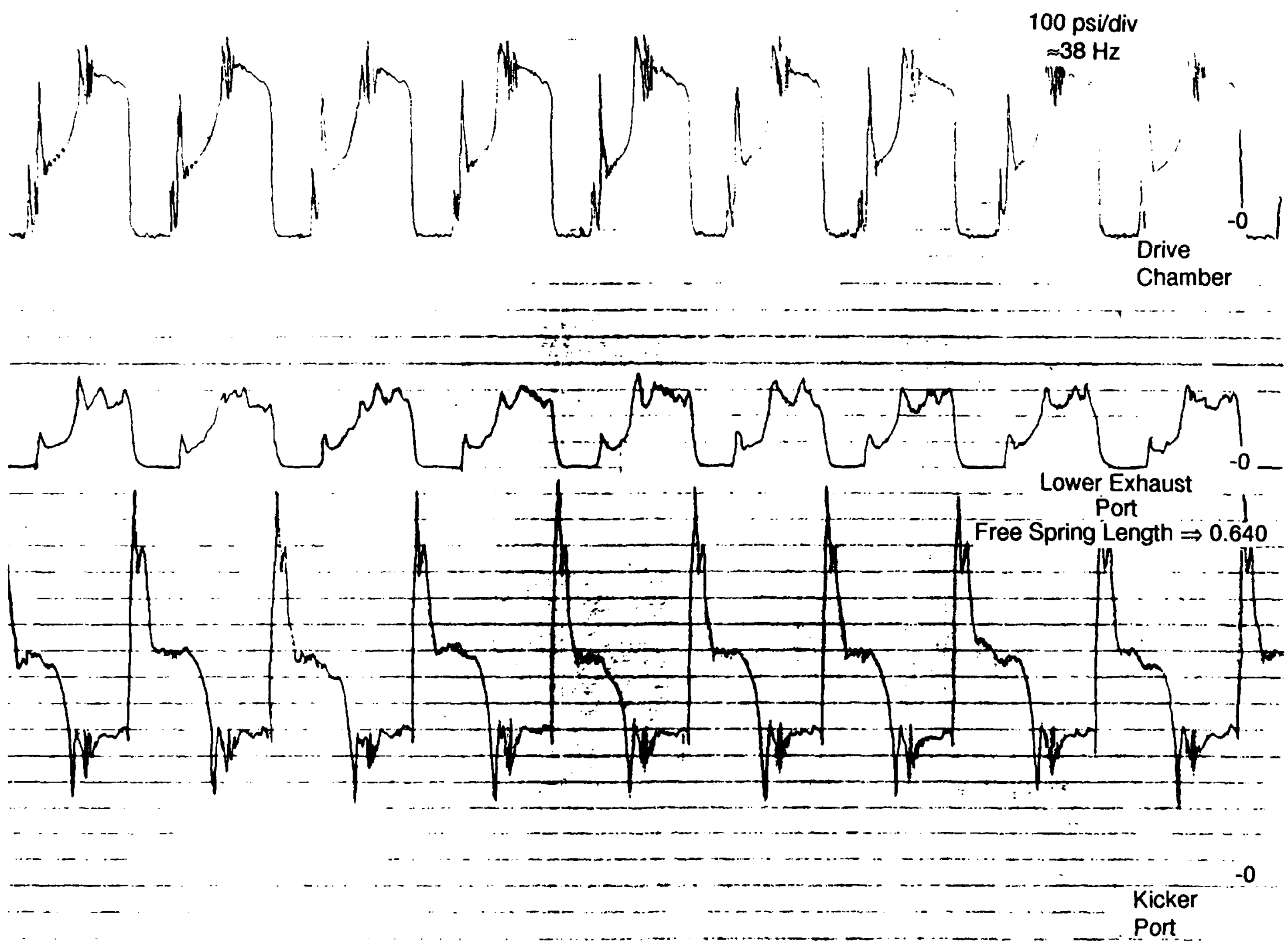


Figure 8. Exhaust pressure follows drive chamber pressure.

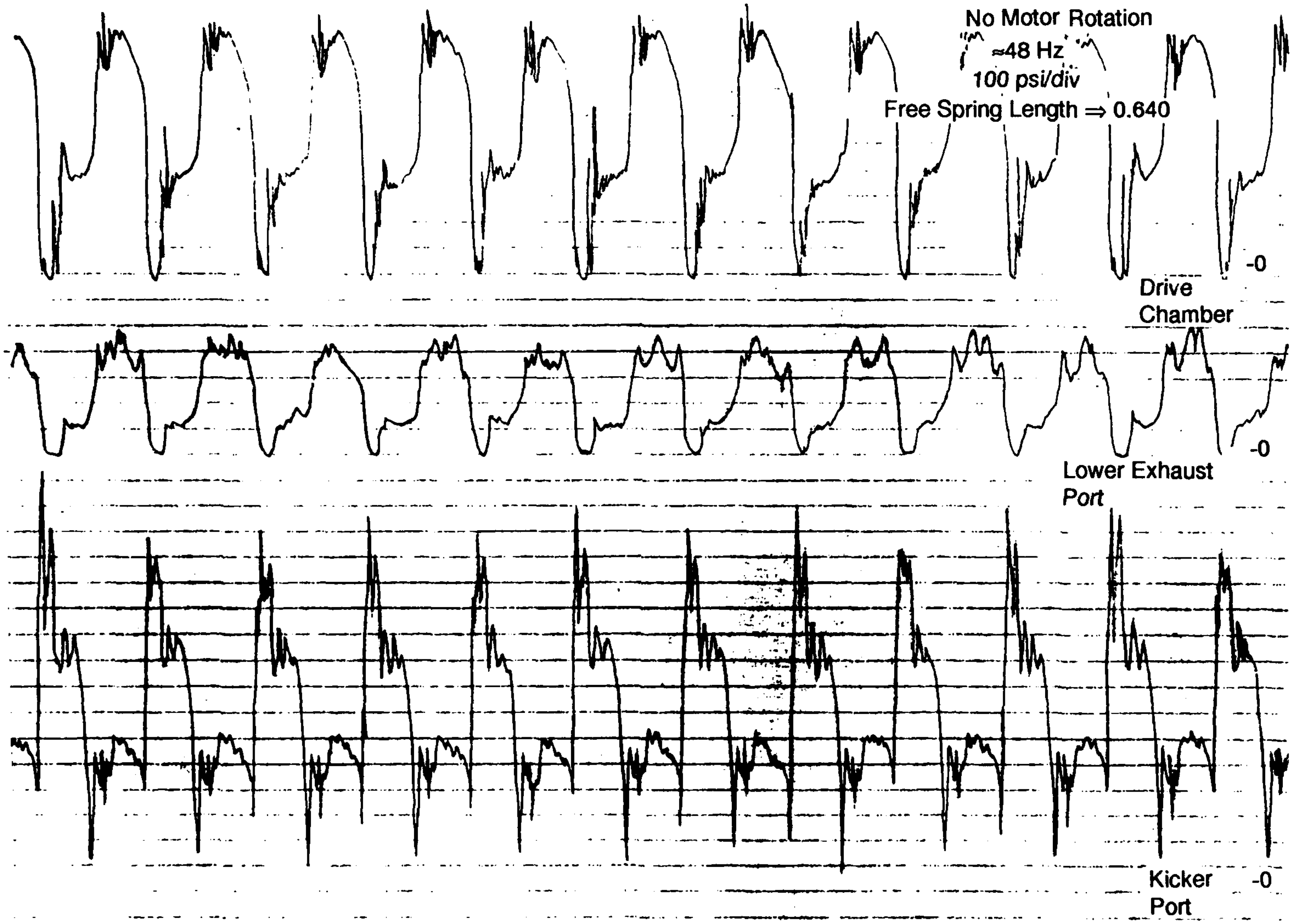


Figure 9. Increased drive chamber pressure.

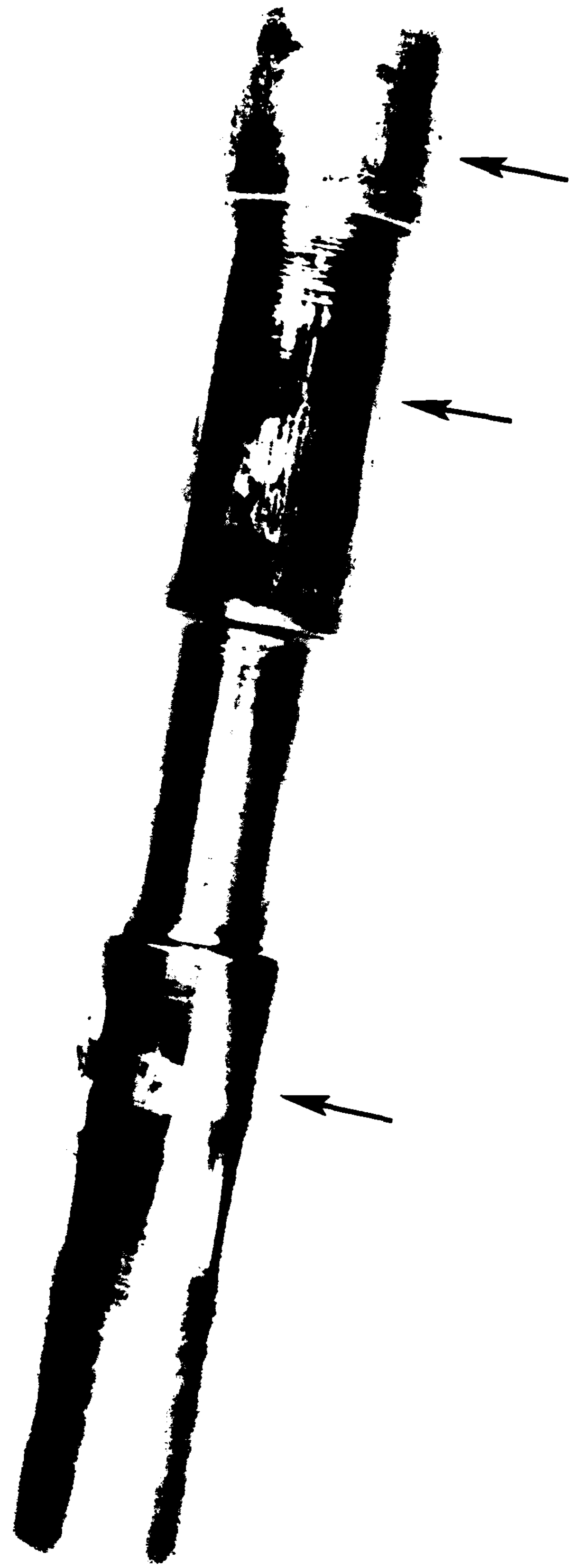


Figure 10. Nedox coating damage to the drive plunger.

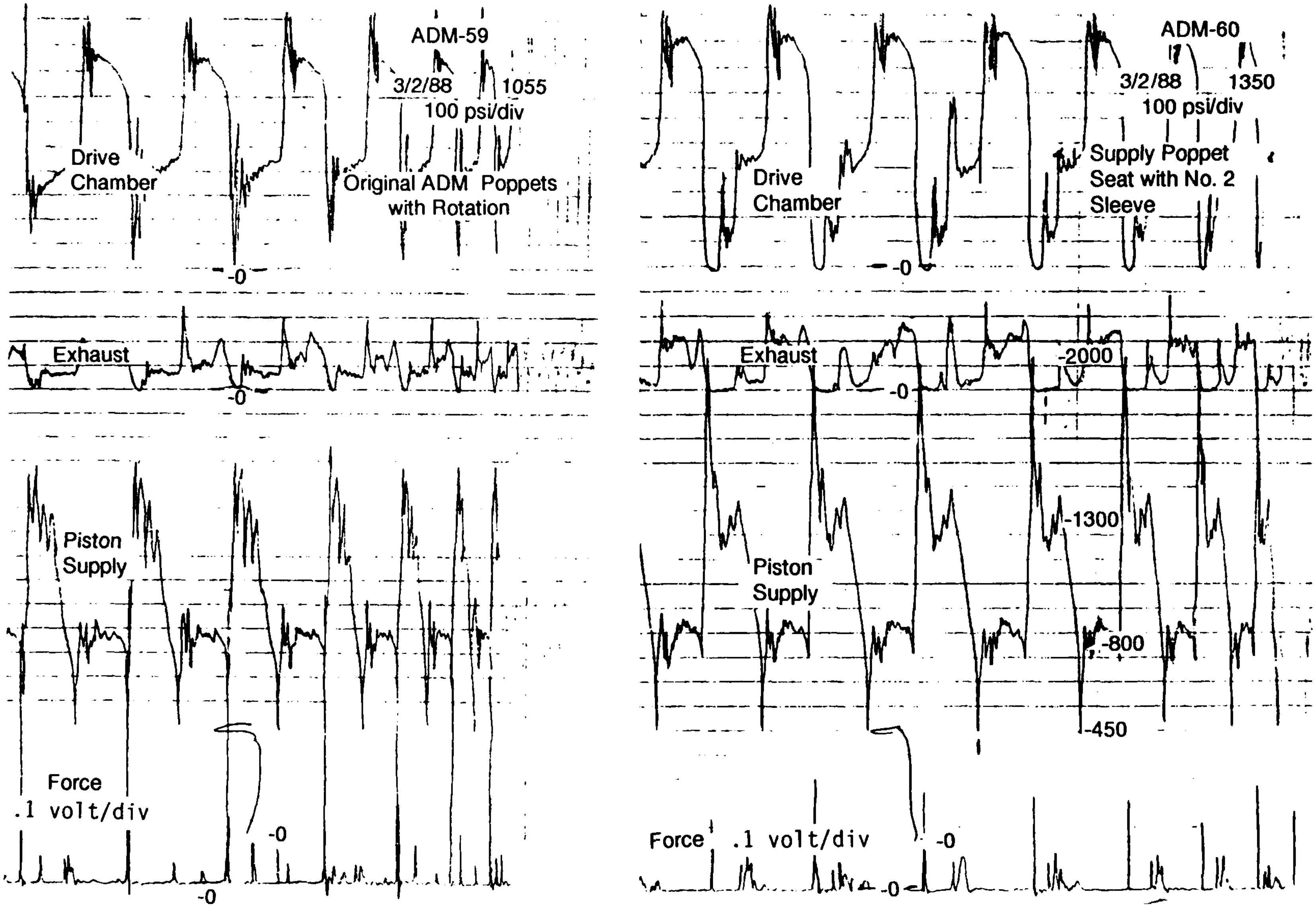


Figure 11. Differences between old and new poppets.

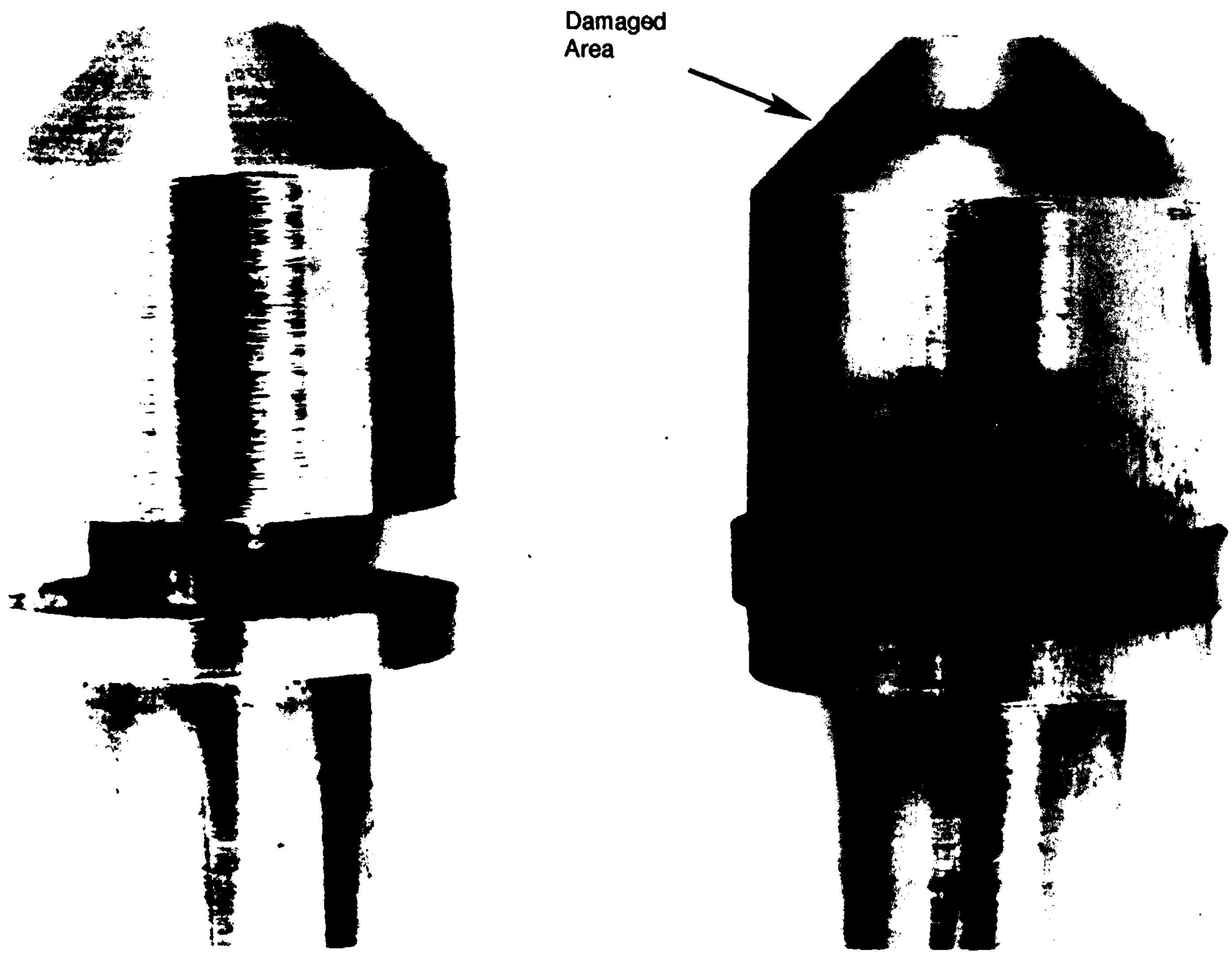


Figure 12. Comparison of new and used supply poppet.

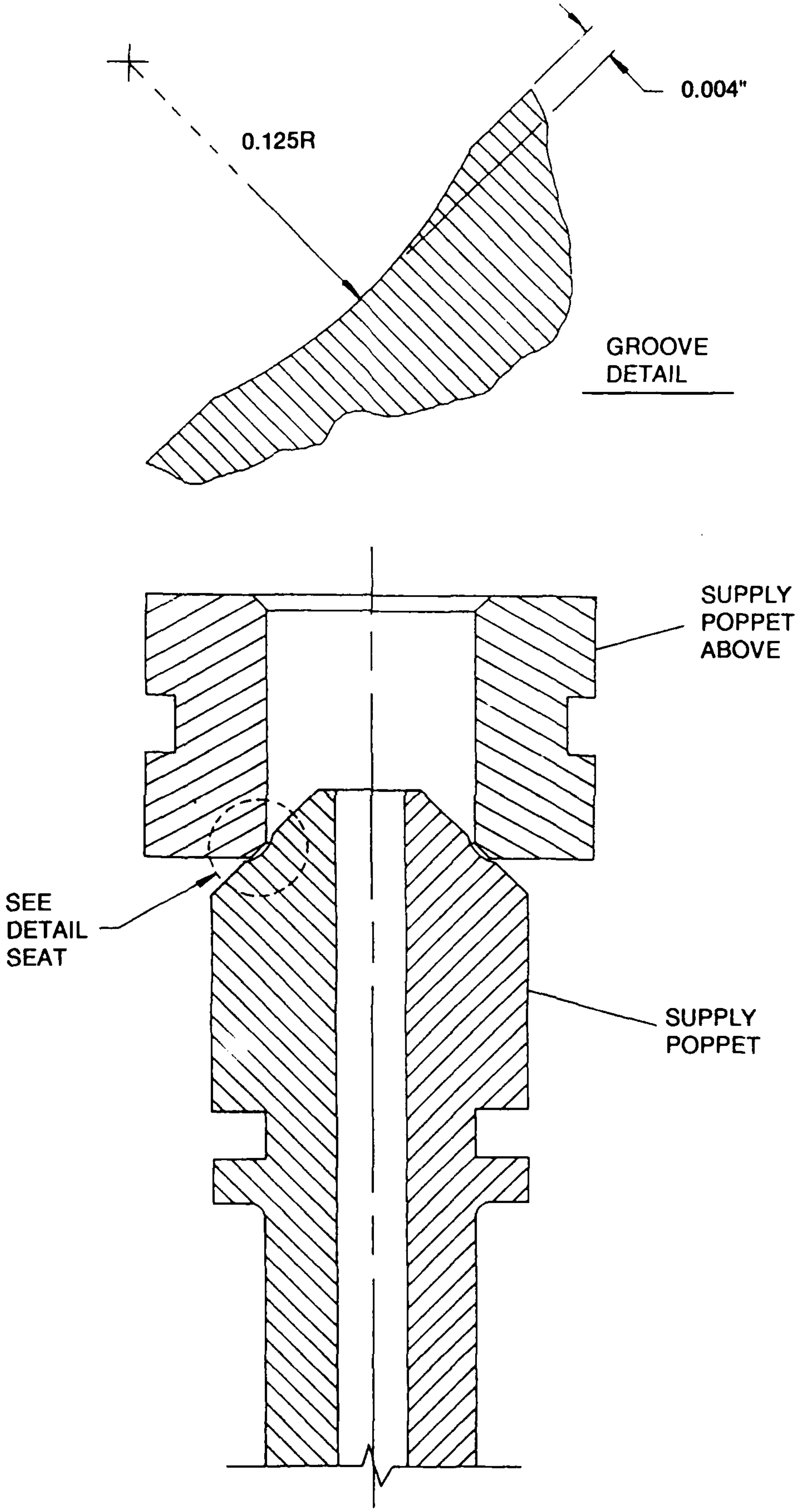


Figure 13. Supply poppet groove profile.

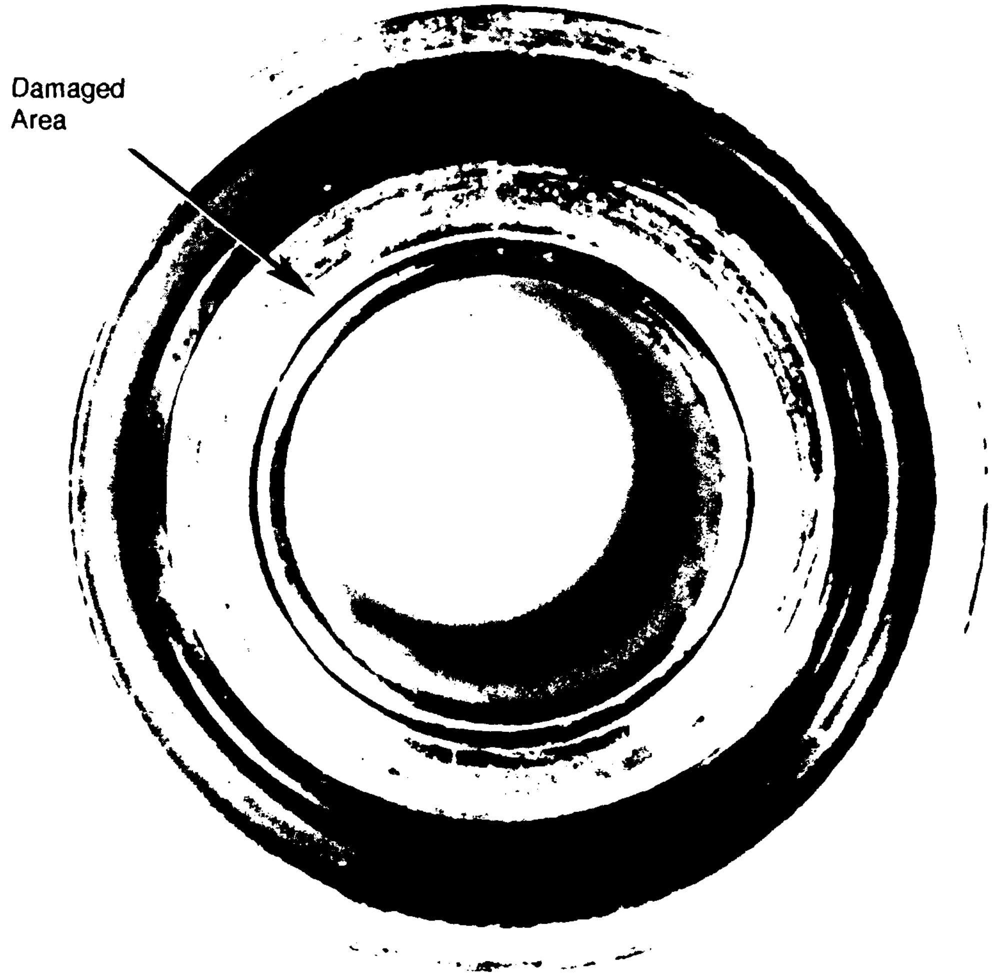
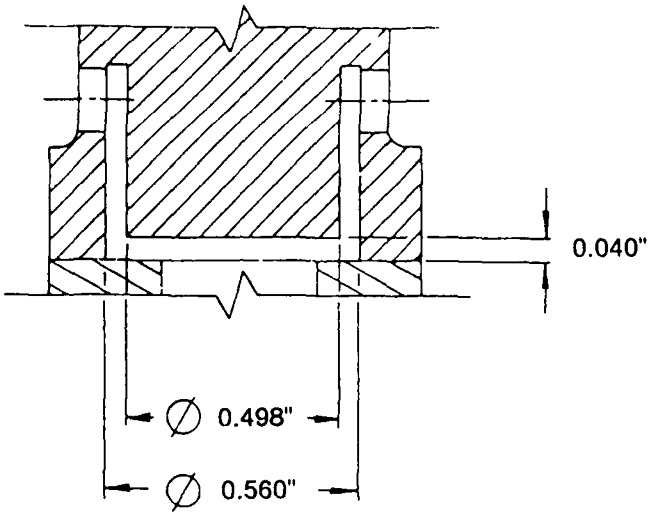


Figure 14. Damaged seat on front plunger sleeve.



ORIFICE DETAILS

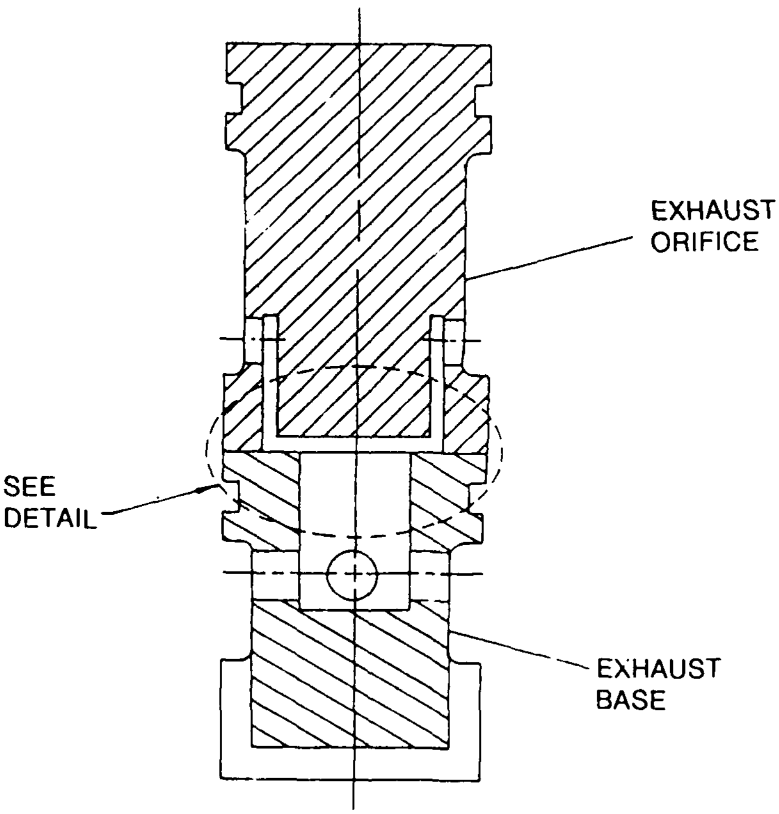


Figure 15. Exhaust orifice design.

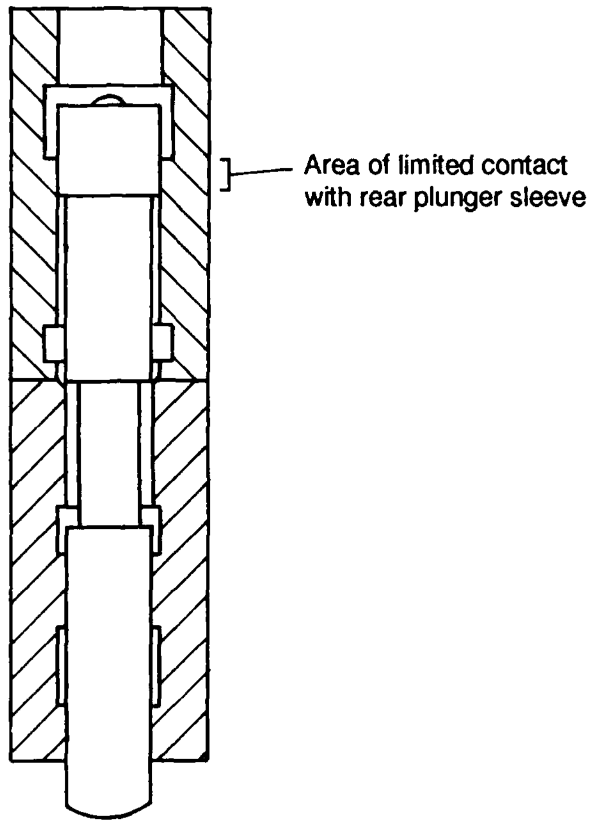


Figure 16. Drive plunger and sleeves layout.

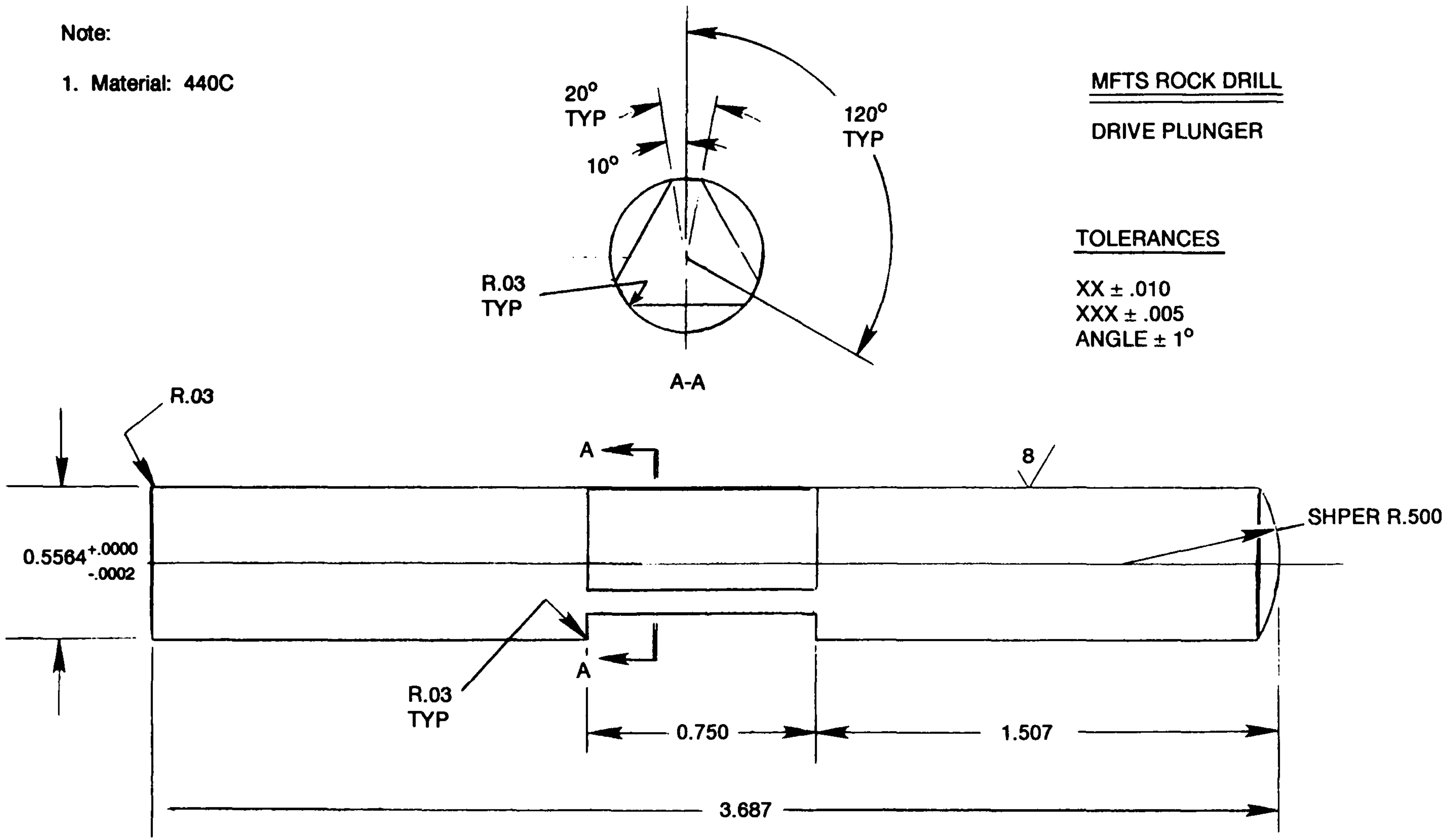


Figure 17. Single diameter drive plunger.

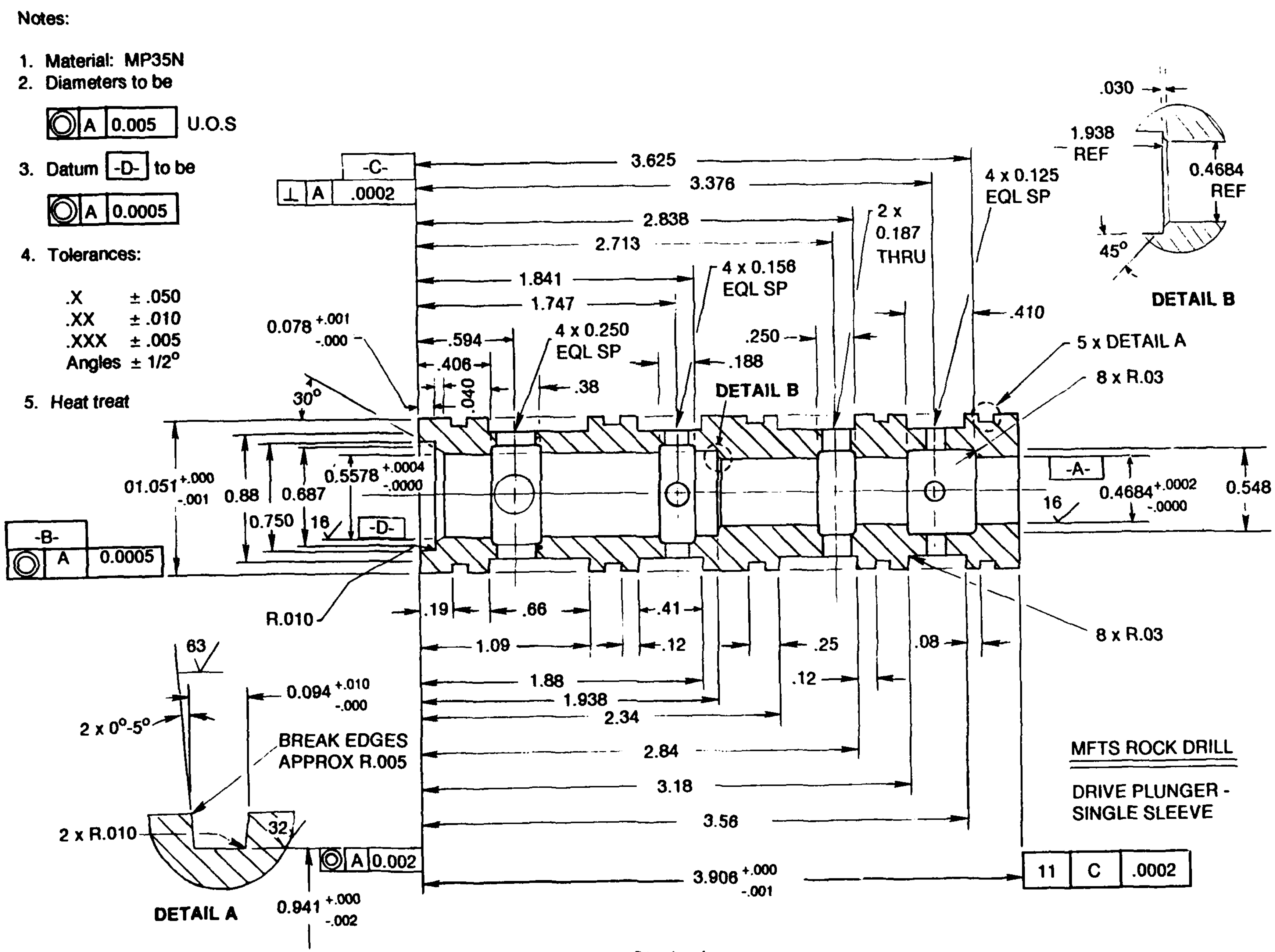
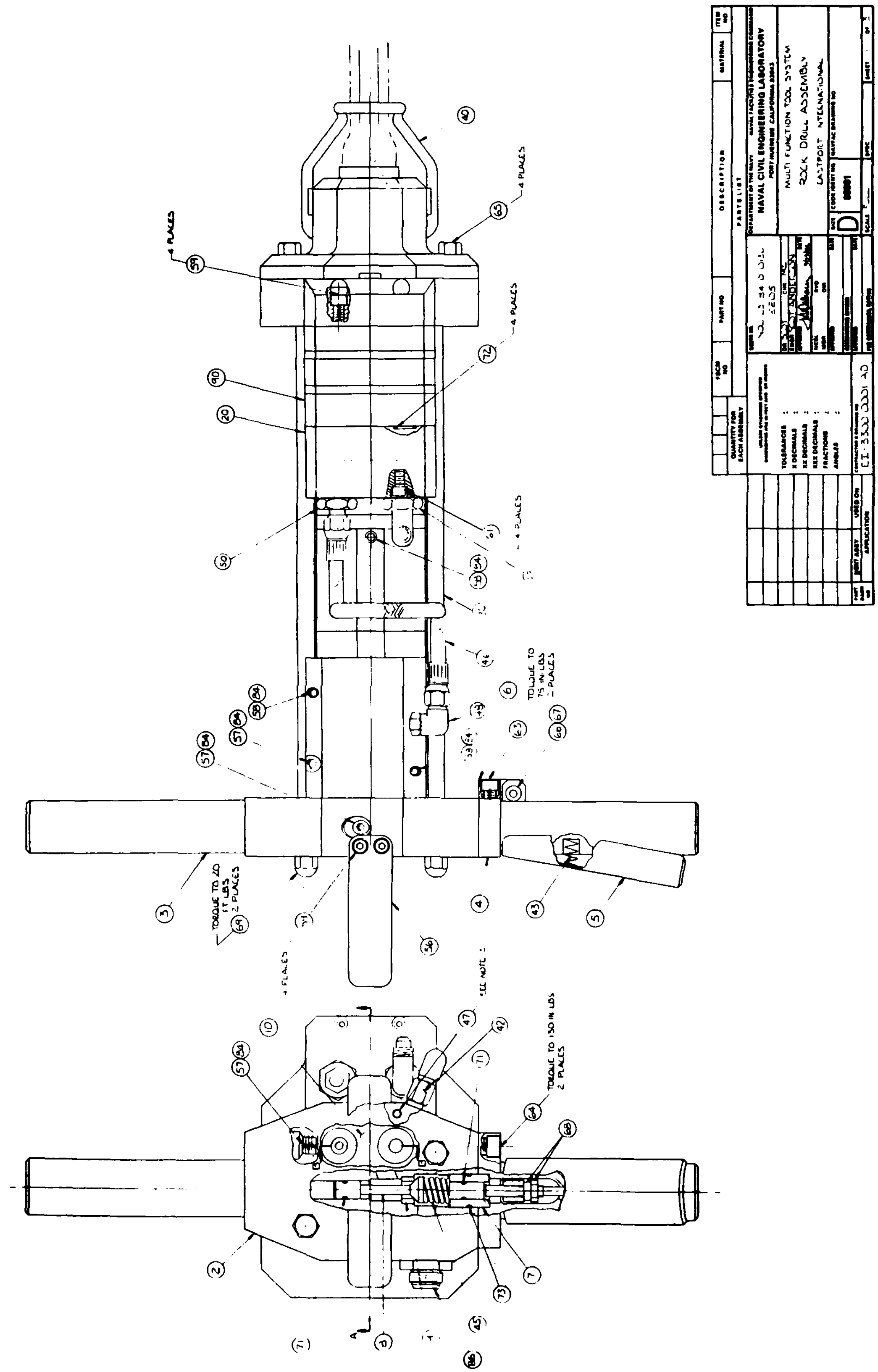
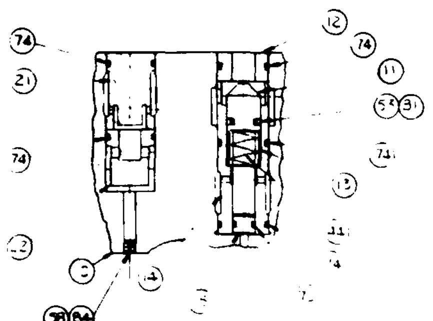


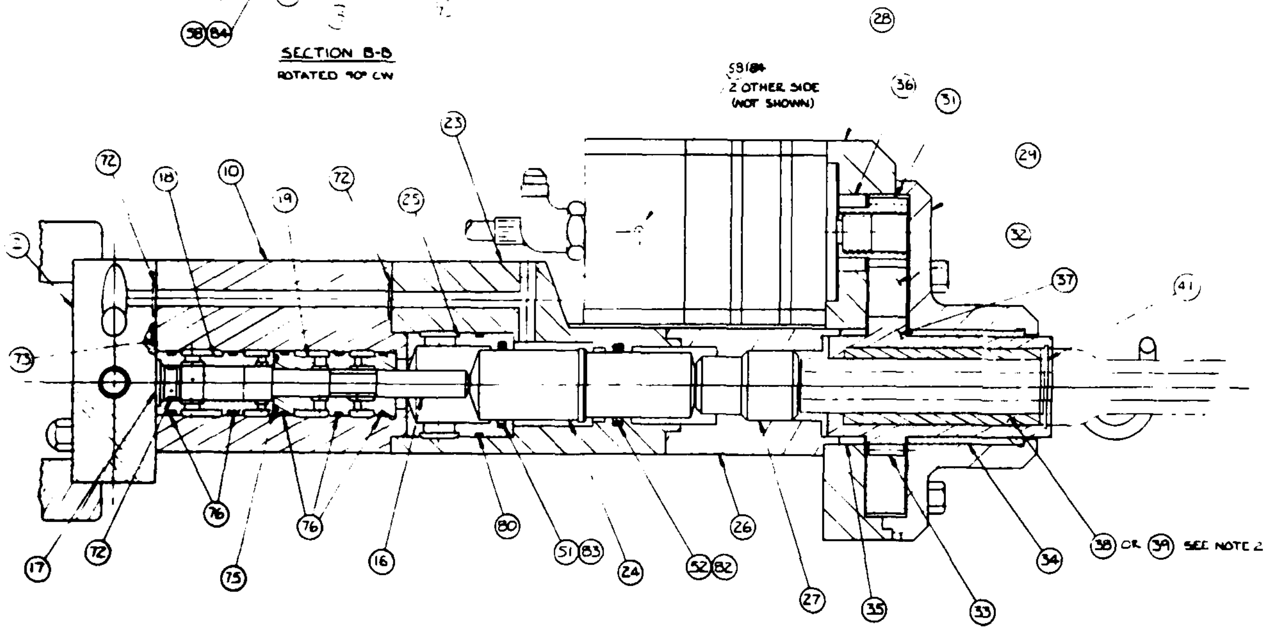
Figure 18. Single sleeve.

Appendix A

FINAL FABRICATION DRAWINGS


4 PLACES
TORQUE TO 20 FT LBS 2 PLACES
TORQUE TO 75 IN LBS 2 PLACES
TORQUE TO 150 IN LBS 2 PLACES
PARTS LIST
DESCRIPTION
MATERIAL
PART NO
QUANTITY FOR EACH ASSEMBLY
TOLERANCES
NAVAL CIVIL ENGINEERING LABORATORY
MULTI FUNCTION TOOL SYSTEM
ROCK DRILL ASSEMBLY
EASTPORT INTERNATIONAL
NEXT ASSY
USED ON
APPLICATION





(CONT. ON SHT 3)

QTY	ITEM NO	PART NO	DESCRIPTION
2	30	EI-3300-0030-DD	TIE BOLT
1	29	EI-3300-0029-DD	FRONT FLANGE
1	28	EI-3300-0028-DD	FRONT HEAD
1	27	EI-3300-0027-DD	ANVIL
1	26	EI-3300-0026-DD	ANVIL HOUSING
1	25	EI-3300-0025-DD	PISTON SLEEVE
1	24	EI-3300-0024-DD	PISTON
1	23	EI-3300-0023-DD	PISTON HOUSING
1	22	EI-3300-0022-DD	EXHAUST BASE
1	21	EI-3300-0021-DD	EXHAUST ORIFICE
1	20	EI-3300-0020-DD	MOTOR PORTING BLOCK
1	19	EI-3300-0019-DD	FRONT PLUNGER SLEEVE
1	18	EI-3300-0018-DD	REAR PLUNGER SLEEVE
1	17	EI-3300-0017-DD	PLUNGER SLEEVE CAP
1	16	EI-3300-0016-DD	DRIVE PLUNGER
1	15	EI-3300-0015-DD	SUPPLY POPPET END PLUG
1	14	EI-3300-0014-DD	SUPPLY POPPET SLEEVE
1	13	EI-3300-0013-DD	SPRING SLEEVE
1	12	EI-3300-0012-DD	SUPPLY POPPET SEAT
1	11	EI-3300-0011-DD	SUPPLY POPPET
1	10	EI-3300-0010-DD	PLUNGER/POPPET HOUSING
1	9	EI-3300-0009-DD	VALVE SEAT
1	8	EI-3300-0008-DD	VALVE PLUNGER
1	7	EI-3300-0007-DD	VALVE PLUNGER SLEEVE
1	6	EI-3300-0006-DD	BRACKET
1	5	EI-3300-0005-DD	LEVER WELDMENT
1	4	EI-3300-0004-DD	HANDLE WELDMENT
1	3	EI-3300-0003-DD	HANDLE
1	2	EI-3300-0002-DD	BACK FLANGE
	1	EI-3300-300 AD	

QUANTITY FOR EACH ASSEMBLY

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN FEET AND/OR INCHES

TOLERANCES: X DECIMALS: XX DECIMALS: XXX DECIMALS: FRACTIONS: ANGLES:

NEXT ASSY | USED ON | APPLICATION

N00123-84 D 0130
2205

SCOT ANDERSON

DEPARTMENT OF THE NAVY — NAVAL FACILITIES ENGINEERING COMMAND
NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME, CALIFORNIA 93043

MULTI FUNCTION TOOL SYSTEM
ROCK DRILL ASSEMBLY
EASTPORT INTERNATIONAL

SIZE D | CODE IDENT NO | NAVFAC DRAWING NO

DRAWING NO 3001-AD

SCALE FULL | SHEET OF

NOTES:

1 INSTALL PLUGS (ITEM 55) IN MOTOR PORTING HOLES A-A AND B-B IN FRONT END HOUSING OF SEAWATER MOTOR (ITEM 90).

2 USE ADAPTERS (ITEM 38 AND 39) TO SHIM DRILL SHANKS AS REQUIRED.

3 PRESS FIT PINS (ITEM 47) INTO PLUNGER/POPPET HOUSING (ITEM [illegible]) 3 PLACES, ANVIL HOUSING (ITEM 26) 2 PLACES, AND FRONT HEAD (ITEM [illegible]) 1 PLACE

QTY	ITEM NO	PART NO	DESCRIPTION
1	90	83-22-06	SEAWATER MOTOR ASSEMBLY
	89		
	88		
	87		
1	86	COMMERCIAL	ADAPTER "PARKER" 0503-10-8C
4	85		HYDRAULIC INSERT PLUG "LEE" PLGK 0503070 (NOT SH[illegible]
AR	84		SST PIPE SEALANT "LOCTITE CORP
1	83		SQUARE RING "MICRODOT" 0275
1	82		0265
1	81		SQUARE RING MICRODOT" 0115
1	80	COMMERCIAL	O-RING "PARKER" 2-128-N674-70
	79		
	78		
	77		
5	76	COMMERCIAL	O-RING "PARKER" 2-021-N674-70
1	75		2-019-N674-70
5	74		2-017-N674-70
2	73		2-016-N674-70
7	72		2-013-N674-70
2	71		2-010-N674-70
1	70		O-RING "PARKER" 2-009-N674-70
2	69		CAP NUT (ACORN) "ALL METAL" 3/8-16UNC-2B SST
2	68		HEX JAM NUT 1/4-20UNC-2B SST
1	67		HEX LOCK NUT "ALL METAL" #10-24UNC-2B SST
1	66		SOC HD SHOULDER SCR "ALL METAL" .246/.248 DIA x .500 LG SST
4	65		HEX HD CAP SCR 5/16-18UNC-2A x 1.25 LG 316 SS
2	64		SOC HD CAP SCR 5/16-18UNC-2A x .63 LG
2	63	COMMERCIAL	1/4-20UNC-2A x .63 LG
	62		
4	61	COMMERCIAL	SOC HD CAP SCR #10-24UNC-2A x .75 LG 316 SS
QTY	ITEM NO.	PART NO	DESCRIPTION

(CONT AT LEFT)

QTY	ITEM NO	PART NO	DESCRIPTION
	60		
8	59	COMMERCIAL	SOC HD CAP SCR #10-24UNC-2A x .875 LG 316 SS
7	58		SOC PRESS PLUG 1/16-27 NPT SST
3	57	COMMERCIAL	SOC PRESS PLUG 1/4-18 NPT SST
1	56	EI-3300-0041-DD	LIFTING HANDLE
	55		
1	54	EI-3300-0040-DD	HOSE WHIP (NOT SHOWN)
1	53	EI-3300-0039-DD	SPLIT SEAL RING
1	52	EI-3300-0038/2-DD	SEAL RING
1	51	EI-3300-0038/1-DD	SEAL RING
1	50	COMMERCIAL	ADAPTER "PARKER" 0503-6-4C
1	49		ADAPTER 90° "PARKER" 2503-6-4C
1	48	COMMERCIAL	ELBOW 90° "PARKER" 2103-2-4C
6	47	COMMERCIAL	DOWEL PIN .1878/.1880 DIA x .75 LG SST
1	46		HOSE, WIRE BRAID SST "PARKER" K9190606444C-8
1	45		SPRING, "ASSOCIATED SPRING" C0600-072-0880S
2	44		C0480-045-1000S
1	43		SPRING, "ASSOCIATED SPRING" C0300-026-0750S
1	42		REDUCING ADAPTER "CAJON" -2-RA-1
1	41	COMMERCIAL	RETAINING RING INTERNAL, IRR # 4000-118
1	40	EI-3300-0037-DD	STEEL RETAINER
1	39	EI-3300-0036/2-DD	HEX ADAPTER, .875
1	38	EI-3300-0036/1-DD	HEX ADAPTER, .750
1	37	EI-3300-0035-DD	HEX CHUCK
1	36	EI-3300-0034/3-DD	BUSHING
1	35	EI-3300-0034/2-DD	
1	34	EI-3300-0034/1-DD	BUSHING
6	33	EI-3300-0033-DD	GEAR PIN
1	32	EI-3300-0032-DD	DRIVEN GEAR
1	31	EI-3300-0031-DD	DRIVE GEAR
QTY	ITEM NO	PART NO	DESCRIPTION

QUANTITY FOR EACH ASSEMBLY

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN FEET AND OR INCHES

TOLERANCES :
X DECIMALS :
XX DECIMALS :
XXX DECIMALS :
FRACTIONS :
ANGLES :

N00123-84-D-0130
ZE05

DR SCOT ANDERSON

DEPARTMENT OF THE NAVY NAVAL FACILITIES ENGINEERING COMMAND
NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME CALIFORNIA 93043

MULTI-FUNCTION TOOL SYSEM
ROCK DRILL ASSEMBLY
EASTPORT INTERNATIONAL

SIZE D CODE IDENT NO 80081 NAVFAC DRAWING NO

EI-3300-0001-AD

SCALE NONE SHEET 3 OF 3

APPLICATION

Appendix B

DRILLING TESTS - HUMAN FACTOR COMMENTS

Field Test, Rock Drill

Record human factors data. (EACH DIVER MUST USE SEPARATE FORM.)

Diver: Tom Conley Date: 6-6-86

Rate the following tool attributes:	Good	Fair	Poor
Trigger mechanism	G	F	P
Comments Excellent, No effort			
Trigger operation	G	F	P
Comments Excellent, works when you want it to not in the way to inadvertantly trigger tool			
Trigger comfort	G	F	P
Comments You don't need to apply any force to act the trigger, you don't even know it's there			
Balance of tool	G	F	P
Comments			
Weight of tool	G	F	P
Comments Tool can be handled easily enough, yet it is heavy enough so you don't need to breakdown on it			
Ease of operation	G	F	P
Comments Aim and shoot			
Water exhaust location	G	F	P
Comments Couldn't feel it in normal operation			

Noise level Low Medium High Too High

Comments High frequency, low amplitude

Do you feel confident that the rock drill would operate and perform its intended task in future operations? If not, please explain.

In clean water as it now exists

Field Test, Rock Drill - Continued

Record human factors data. (EACH DIVER MUST USE SEPARATE FORM.)

Diver: Ron Erich Date: 6-6-86

Rate the following tool attributes:	Good	Fair	Poor
Trigger mechanism	G	F	P
Comments: Difficulty lifting without turning on trigger			
Trigger operation	G	F	P
Comments			
Trigger comfort	G	F	P
Comments			
Balance of tool	G	F	P
Comments			
Weight of tool	G	F	P
Comments: Somewhat light, tendency to want to help it; S-8 # heavier			
Ease of operation	G	F	P
Comments			
Water exhaust location	G	F	P
Comments			

Noise level Low Medium High Too High

Comments Compared to other drills

Do you feel confident that the rock drill would operate and perform its intended task in future operations? If not, please explain.

Fine to 1 1/2" - too low power for larger holes.

Swivel or 90° straight up on hose so hose won't interfere with diver - could be used right handed

Field Test, Rock Drill - Continued

Record human factors data. (EACH DIVER MUST USE SEPARATE FORM.)

Diver: Steve Koepenick Date: 6-6-86

Rate the following tool attributes:	Good	Fair	Poor	
Trigger mechanism	G	F	P	
Comments				
Trigger operation	G	F	P	
Comments Except when attempting to remove drill that is "stuck"				
Trigger comfort	G	F	P	
Comments				
Balance of tool	G	F	P	
Comments Shank too long/How about shorter "starter" shank				
Weight of tool	G	F	P	
Comments Heavier might be better				
Ease of operation	G	F	P	
Comments Once principle is understood				
Water exhaust location	G	F	P	
Comments Clouds up water/can exhaust be directed to direct "cloud" away from operation?				
Noise level	Low	Medium	High	Too High
Comments				

Do you feel confident that the rock drill would operate and perform its intended task in future operations? If not, please explain.

Not sure I should state due to lack of experience. What about more torque? There were times when drill was stuck but I did not realize.

6 June 86

3/4" Diameter Twist Bit; Tom Conley's and Ron Erich's comments

- no balance problems - light enough

- seems heavy enough for good handling in surf zone

- heavier than HD-20

- restart hole - can't see bit - motor cuts down on visibility

- starting hole easy - (3/4)

- motor and inlet same side?

hose doesn't pose problem, no swivel
bring hose out through handle

- comparison w/oil - oil feels like it is going into hole - lack of this sensation H_2O

- feels like elastic inpact wrench

- noise - level okay above tool but at tool level - irritating

1-1/2" Diameter Cross Bit

Ron Erich's comments: Feels same as small bits, needs more power to motor - bit stalled frequently in deeper holes. Use starter bit made for starting large hole in rock. Drill is not too heavy - doesn't overpower diver, easy to maneuver. Could tire diver with frequent stalling.

Appendix C

SAFETY ANALYSIS

ROCK DRILL HAZARD ANALYSIS

The hazards associated with diver operation of the Multi-Function Tool System rock drill are the hazards normally associated with diving plus additional problems caused by tool operation. This analysis focuses on tool operation for the purpose of establishing requirements for operator safety devices. Hazards for operation of the MFTS rock drill have been classified according to their severity, probability of occurring, and risk assessment code (RAC). This information is followed by a brief discussion for preventative action. Potential hazards in operating the MFTS rock drill are:

Table C-1. Preliminary Hazards Analysis

Fault	Effect	Severity	Probability	RAC	Recommended Action
		Damage to Tool			
Improper Assembly	Tool Inoperable	4	C	5	Provide training using O&M manual
Dropped Tool	Tool Inoperable	3	B	3	Provide training on handling
		Injury to Operator			
Dropped Tool	Foot Injury	3	B	3	Provide training on handling; wear safety shoes topside
Noise Exposure	Hearing Damage	3	B	3	Provide training using O&M manual; limit exposure
Water on Deck	Slip and Fall	3	A	2	Restrict topside operation; wear deck shoes
Excessive Pressure	Burst Hose	3	D	5	Provide training using O&M manual; safety relief valve
Unplanned Actuation	Dropped Tool	3	B	3	Provide training on proper handling; auxiliary handle; de-energize tool during transit
Freeing a Stuck Bit	Loss of Balance	3	B	3	Provide training on proper removal procedures

Description of the hazard rankings are:

SEVERITY

1 Catastrophic: Death or system loss.

2 Critical: Severe injury or major system damage.

3 Marginal: Minor injury or minor system damage.

4 Negligible: Less than minor injury or system damage.

PROBABILITY

A Likely to occur frequently

B Will occur several times

C Likely to occur sometime

D Unlikely byt possible to occur

E Assumed not to be experienced

RISK ASSESSMENT CODE (RAC)

1 Elimination or positive control of the fault causing this hazard is imperative.

2 Elimination or positive control of the fault causing this hazard is highly desirable.

3 If the fault producing this hazard cannot be eliminated, some control over the effect should be exercised.

4 Minimal effort should be expended on the elimination or control of the fault causing this hazard.

5 No effort need be expended on correcting this fault.

Appendix D

DESIGN REVIEW CONSULTANT REPORTS

Krasnoff Consulting Associates
107 Kingsway Common Princeton, N.J. 08540 (609) 497-0895

Mechanical Engineering Specialists

SEAWATER POWERED ROCKDRILL

DESIGN REVIEW

Prepared for

EASTPORT INTERNATIONAL
Pacific Marine Systems Division

January 28, 1988

SUMMARY

A review of the Pre-Production Prototype detail design package was undertaken with emphasis on basic hydro-mechanical cycle features. Several design deficiencies which produce poor cycle performance and could lead to malfunction were uncovered. These deficiencies and the hardware modifications required for their elimination are discussed in this report.

Fortunately, the required hardware modifications are not extensive and it is expected that the Pre-Production Prototype will meet the drill rate requirements of the Test and Evaluation Master Plan. On the other hand, the results of the design review demonstrate that the exhaust valve of the PPP is a most unreliable element of the rock drill. This stems from the fact that i) its proper functioning depends on a sensitive balance of hydro-mechanical forces and ii) dirt particles from the ambient will be present in the clearance area of the poppet - sleeve assembly. These particles will easily upset the required balance of forces.

An alternative valve concept is suggested for the Production Seawater Powered Rock Drill. This alternative eliminates the exhaust poppet assembly entirely and maintains the basic reliability of the supply poppet design as well as that of the basic kicker port rock drill cycle. In addition, it will lead to a lower cost of manufacture due to the elimination of several expensive parts and a simplification of the plunger-poppet housing.

I Introduction. Scope of the Design Review.

The brief design review summarized in this report was undertaken in view of the poor initial operational reliability of the pre-production prototype rock drill. Therefore, emphasis was placed on the basic actuator cycle components such as the exhaust valve, supply valve, plunger and piston assemblies. The Appendix to this text presents the results of hydrodynamic calculations which indicate some serious problem areas. These results, which are based on the dimensions and tolerances called for in the detail drawings, are discussed in Section II along with recommendations for required hardware modifications.

During the course of examining the detail drawings, material selections, coatings and mechanical design features were noted, and these are discussed in Section III. Finally, in Section IV the drill system design concept is considered and recommendations for changes are made. These changes could lead to benefits relating to cost of manufacture, operational reliability and a decreased length of the drill package.

II Review of the Cycle Hydro-Mechanical Design.

2.1 Plunger - Sleeve Assembly.

Section A-I of the Appendix shows that the plunger-rear sleeve diametral clearance can be smaller than the clearance in the front sleeve. Since the front end must be the bearing area this could cause a problem with excessive wear rates or scuffing due to wobbling of the plunger during its reciprocating motion. The nominal clearances are rather large (about twice those called for in the Advanced Development Model), and tilting of driven reciprocating elements is known to cause problems.

The large clearances also lead to excessive leakage flows from the kicker port gallery to the ambient through the exhaust valve and through the vented region in the rear piston housing. The former leak rate could produce a malfunction in the valve

action as a secondary effect. (The calculation results presented below do not indicate a problem here. Thus, the effect on valve action would be secondary in that it could aggrevate the effects described in Sections 2.2 and 2.3.)

2.2 Exhaust Valve Assembly.

The possibility of a premature opening of the exhaust valve during the drive stroke is demonstrated by the calculation results of Section A-II. While this is not a likely event it is recommended that the valve seat I.D. be opened slightly to prevent its occurence (see item 4. of Section A-II).

The maximum clearance around the valve stem is 0.0018 in. and this is much too large. Experience with valve elements in water powered drills suggests that this valve design will admit tilting during the valve motion, and this will produce high wear rates and frictional retarding forces. In this connection it is noted that both ends of the poppet are vented to the ambient. Thus, dirt particles will be ingested into the valve assembly during the stroking of the valve. The best way to prevent valve sticking is to use a small clearance (0.0005 in. nominal) and machine anti-lock grooves into the valve stem. These grooves, which are often thought to prevent "hydraulic lock", really prevent sticking by allowing dirt particles to collect in the grooves.

The exhaust poppet design concept must be considered a mistake because, as noted in A-II, it is a sensitive and unreliable element. This was evident from the results of early tests of the ADM. Therefore, an alternate design concept should be given serious consideration for the production drill. The concluding section of this report presents an alternative which should be considered in confidence as proprietary information.

2.3 Supply Valve Assembly.

Section A-III shows that the kicker port gallery is closed off from the supply gallery when the plunger is in its full forward position. Similarly, it is cut off from the exhaust

gallery when the plunger is near its top dead center position. The schematic of Figure 1 demonstrates the problem and the solution: the length of the kicker port gallery must be increased as shown. Reliable poppet action cannot be expected without making this simple modification.

Extensive leak flow calculations were made to determine the kicker port pressures at various stages of the plunger motion. The results presented in A-III demonstrate that the kicker action will be as desired even though the plunger clearance is larger than as specified in the ADM design. Thus, with the specified change in the length of kicker port gallery, supply poppet action should be reliable, at least for operation at full system pressure. The stipulation of full system is made because it is possible that excessive poppet seal friction in combination with a large spring force can prevent the poppet from cracking open at low supply pressures. This is indicated by the calculation of item 2. of A-III. The situation would be alleviated by a slight increase in the poppet seat I.D. In any event, seal friction should be checked on assembly to verify that o-ring tolerance does not result in excessive squeeze and static seal friction.

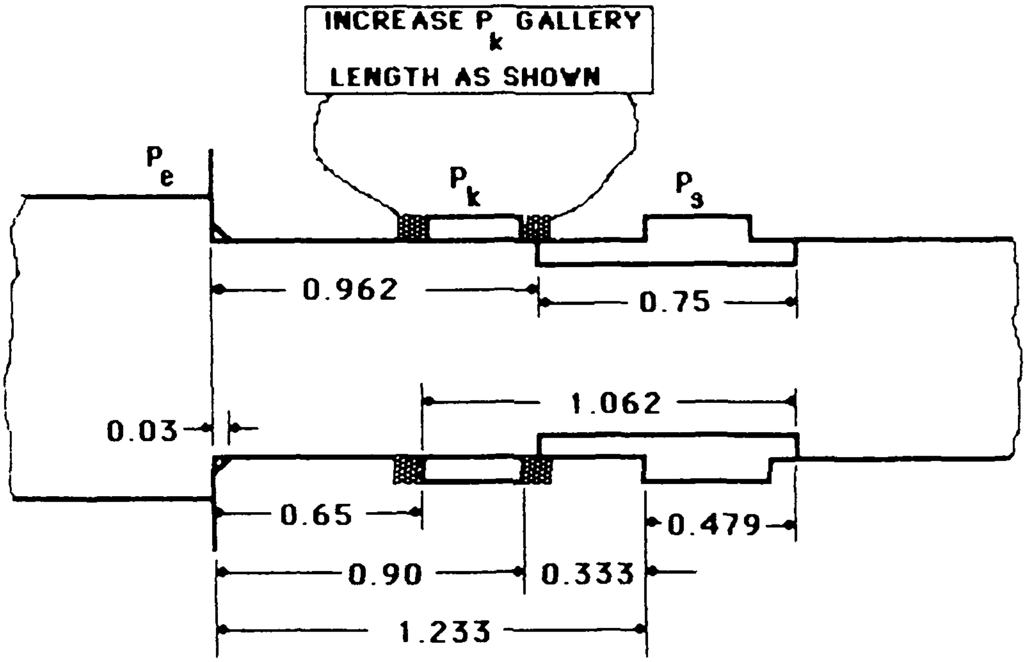
2.4 Plunger - Piston Axial Travel Limits.

Dimensional checks of the detail drawings show that the plunger travel between the front and rear cushions is only 0.967 in. This is less than the original design point piston stroke and, since the pressure drop during the drive stroke will be large due to the lack of a supply line accumulator, the desired blow energy will not be achieved. In addition, as noted in A-IV, the kicker port signal for the poppet to open occurs when the plunger has traveled 0.932 in. from the full forward position. This means that the plunger will always enter the rear cushion during normal operation.

A dimensional check of the piston - housing assembly shows that the piston will tend to impact the front end of the plunger-poppet housing. Failure of the plunger - poppet housing will be the result.

FIGURE 1

CRITICAL DIMENSIONS OF PLUNGER - SLEEVE ASSEMBLY. PLUNGER FULL FORWARD POSITION.



EXTENDING SLOT LENGTH IN SHADED AREA ON LEFT ALLOWS P_k COMMUNICATION WITH P_e AS PLUNGER APPROACHES FULL STROKE POSITION. THIS GUARANTEES THAT THE SUPPLY POPPET OPENS FULLY TO ESTABLISH THE DRIVE STROKE.

IN POSITION SHOWN P_s DOES NOT COMMUNICATE WITH P_k. THIS WILL LEAD TO CAVITATION IN THE SUPPLY POPPET GALLERY BECAUSE THE POPPET IS CLOSING AND REQUIRES A CONTINUOUS SUPPLY OF FLUID UNTIL IT IS SEATED.

The design package does not admit any simple means of increasing the stroke limits of the plunger and piston. However, details of the PPP assembly should be examined to determine how small increases could be achieved. Perhaps there is a way to increase the piston stroke limit by a tenth of an inch or so. If necessary the piston sleeve could be modified to provide a cushion as the piston comes over top dead center. Also, if necessary, the return stroke could be decreased by increasing the length of the plunger kicker port cutout (on the left side as seen in Figure 1). This would produce an earlier opening of the supply poppet.

2.5 Piston Housing -Piston Sleeve Assembly.

An oversight in the original ADM design has been repeated in the PPP design as indicated in Figure 2. The annular gallery which is necessary for proper venting of the rear piston face is much too small. As indicated in A-V this will have serious consequences on the net forces which accelerate the piston. Fortunately there is sufficient material in both the housing and the sleeve to correct this deficiency.

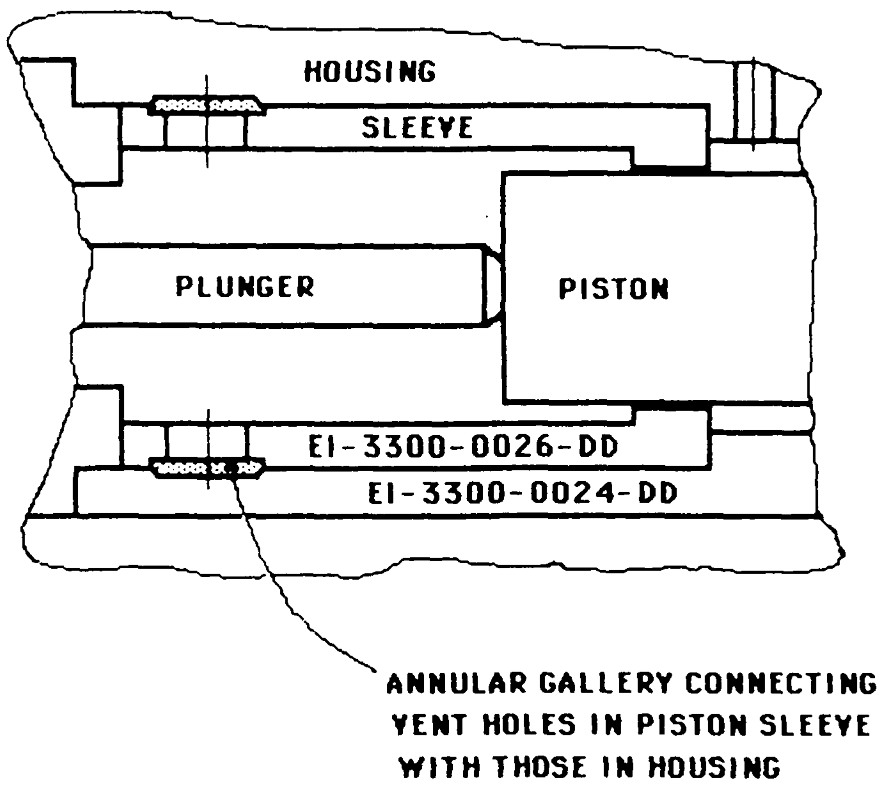
III Mechanical Design.

3.1 Materials.

The supply and exhaust poppets seat with high velocities so impact stresses will be high. In oil powered drills items such as poppets and seats would make use of through hardened tool steels. The use of Nitronic 50 - 30% C.R. for the supply poppet and Nitronic 60 for the supply poppet sleeve, for example, will likely lead to rapid working of the seat and operational failure. Similarly, the wear rate of the Nitronic 60 sleeve will be high. The ADM design called for cold worked SCF stainless for both the poppet and its seat and Stellite for the sleeve to achieve long life. Materials choices which approach the ADM properties should be used for the poppets, poppet sleeves and poppet seats if at all possible. The same is true for the plunger and the plunger sleeves.

FIGURE 2

PISTON SLEEVE - PISTON HOUSING ASSEMBLY



3.2 Coatings.

The use of "NEDOX CR+" from General Magnaplate will have to stand the test of time. Experience with commercial plating dictates that suppliers' claims for bond strength and porosity are often not realized and new plating tests often meet with early failure. During the ADM design effort the best coating for use with water power was CVD-TiC. It must be noted, however, that good results with CVD-TiC were obtained with careful filtration of the working fluid. Dirt from the seawater pump system or the ambient could lead to short life of this coating.

In recent years Ingersoll-Rand in collaboration with the Chamber of Mines of South Africa has continued its long term investigation of coatings. Krasnoff Consulting Associates must consider recent information on this subject as confidential. Therefore, it is suggested that Eastport International contact Ingersoll-Rand to determine their recent experiences. (The proper contact is Mr. R. Lyon: Ingersoll-Rand Rock Drill Division, (703) 362-3321, Ext. 498.)

3.3 Mechanical Design Details.

An item of concern which was noticed during the review of the basic design appears on Drawing EI-3300-0020-DD. The maximum interference fit of the insert in the front plunger sleeve is 0.003 in. This could cause problems with distortion during the press fit operation since the plunger-sleeve clearance can be only 0.0013 in. according to the design detail tolerances.

This is one item of many which may cause assembly or manufacturing problems and which have not been considered in this review. Such items as the stack-up of axial tolerances should be addressed, especially in view of the limited strokes of the plunger and piston. Similarly, proper tapered lead-ins should be used to prevent static o-ring failure during assembly, and the details of o-ring grooves and o-ring squeeze should follow standard practice to insure proper sealing. This point is noted here because a common experience is that a water powered rock drill malfunctions due to static o-ring failure, and this is the last item considered when trouble shooting. In this

connection it is noted that the plunger-poppet housing drawing (EI-3300-0011-DD) does not call out any dimensions on the bore undercut edges. A typical shop could easily supply the housing with sharp edges here. It is also noted, for example, that the tolerancing of the supply poppet bore in the housing and the sleeve o-ring grooves admit a minimum radial clearance between the housing and the groove I.D. of 0.056 in. With the seal groove width of 0.094 in., and considering the typical loose tolerances of o-ring seals, this detail could cause assembly problems.

IV Drill System Considerations.

4.1 Flushing and the Anvil.

The PPP design makes no provision for flushing chips out of the hole during drilling. In the long run this is a mistake. It may be all right for drilling holes only a few diameters long, but drilling rate will become very low for deeper holes. With no flushing the chips remain at the bottom of the hole to be crushed into fine particles. Ultimately, drilling rate will become nil and the rotation motor will become overloaded.

The use of the anvil in the PPP design serves only to produce a small loss in the blow energy transfered to the bit. The anvil was incorporated into the ADM design to facilitate flushing with unmodified, commercially available drill steels. Reference to the Rock Drill Assembly (EI-3300-0001-DD) will indicate the benefit of eliminating the anvil in a future production drill system. That is, it will result in a shorter drill with longer allowable stroke for the plunger and piston. Therefore, it is recommended that the anvil be eliminated in the production drill design. However, flushing should be incorporated into the design via cross holes in the drill steel as shown in the PPP hydraulic circuit drawing.

4.2 Pre-Production Prototype Tests.

As regards future testing of the PPP it is recommended that flushing be added before attempting to drill deep holes. This should follow after the establishment of proper and reasonably reliable cycle operation. To achieve the latter the

modifications of Section II should be made. In addition, the drill assembly should be considered again to determine if the piston and plunger stroke clearances can be increased. Reference to the assembly drawing suggests that this can be accomplished by decreasing the length of the anvil to lower the impact point. Alternately, the anvil can be eliminated entirely if a drill steel with a long shank is used. In the latter case the anvil housing would be modified to provide a bearing for a modified drill steel. In either case the plunger-sleeve assembly would be modified to accommodate the displaced impact point and care would be taken to keep the piston out of its front cushion while drilling.

4.3 The Double Poppet Design Concept.

During the early tests of the ADM it was evident that the exhaust valve was malfunctioning. Thus, clearances have been opened up and the exhaust flow is now ported to the ambient rather than to the drill steel. However, it is apparent that the doubble poppet design concept was a poor choice for a seawater powered rock drill. Reliability of the exhaust poppet is poor because its design must be a delicate balance between i) a low spring force to allow quick closing <u>after</u> the supply poppet opens and ii) a high spring force to permit opening <u>after</u> the supply poppet has closed. Also, it must be noted that the opening of the exhaust poppet can begin only after the supply poppet has closed <u>and</u> the drive chamber pressure has dropped due to leakage around the head of the drive plunger. The delay in the start of the return stroke can be substantial due to this feature, and this aggrevates the supply line pulsations and cuts down on the drill impact frequency and power output.

Another very undesirable feature of the double poppet design is the surge of flow through the drive chamber when the supply poppet opens. The detail design of the exhaust poppet minimizes this surge in terms of the flow loss, but it cannot avoid a short term high speed flow past the exhaust poppet. The presence of this high flow speed in the PPP design can be expected to produce a problem with cavitation erosion in the area of the exhaust poppet seat, especially since the poppet is now ported to the ambient.

4.3.1 A Design Alternative.

The poor reliability of the exhaust poppet was recognized after the first tests of the ADM. Therefore, design concepting was undertaken and this led to a patent invention disclosure of a hybrid poppet-spool valve. This invention disclosure has been presented to NCEL in confidence, and it is now being transmitted to Eastport International under separate cover with the same request for strict confidence.

The new valve makes use of a supply poppet to insure against a large leak loss during the relatively long return stroke of the kicker port cycle. However, it combines the exhaust porting in a single element in such a manner that none of the deficiencies of the PPP design are present. The poppet element has the same basic geometry as the supply poppet of the PPP and, thus, requires only one valve sleeve and seat. The complete exhaust valve assembly (including Drawings EI-3300-0021-DD through -0023-DD and the exhaust poppet spring) is eliminated and the poppet-plunger housing is simplified. The cost benefit is clear, and reliability will be increased significantly since a number of unreliable parts are eliminated. As regards reliability, it is pointed out that the alternative valve has reciprocating end faces just like those of the PPP supply poppet. Thus, unlike the PPP exhaust valve, it does not pump ambient fluid (dirt) into the valve clearances. In the long run this pumping action would be the most serious deficiency of the PPP exhaust poppet.

Eastport International and NCEL are urged to give serious consideration to the new valve concept for the future Production Seawater Powered Rock Drill. Prior to any final design package one of the present PPP's could be modified to test and optimize the detail design.

APPENDIX

Design Review Calculations

and Dimensional Checks

A-1 Plunger-Sleeve Diameters

Detail drawings EI-3300-0017, -0019, -0020 -DD specify the following dimensions and tolerances:

	Sleeve	Plunger
Rear	D= .5578 (+.0004/-0)	.5552 (+0/-.0002)
Front	D= .4684 (+.0002/-0)	.4662 (+0/-.0002)
		Coating: .0007-.0009

Thus, the diametral clearances around the plunger are:

Rear sleeve max. =.5582
Plunger min. = .5557 Max. Diam. clear. = .0025

Rear sleeve min. = .5578
Plunger max. = .5561 Min. Diam. clear. = .0017

Front sleeve max. = .4686
Plunger Min. = .4667 Max. Diam. clear. = .0019

Front sleeve min. = .4684
Plunger Max. = .4671 Min. Diam. clear. = .0013

1. The max. clear. of .0019 in the front sleeve in combination with the min. of .0017 in the rear could cause a problem. Best check actual hardware and modify as necessary to keep front clearance less than rear clearance.

2. Front max. clear. of .0019 has clearance leak flow area of $\pi(.4667)(.0019/2)$ or 0.00139 in^2. This is the equivalent of a 0.042 in. diameter hole. The effect on porting functions must be evaluated, but for now it would be wise to tighten the front clearances to avoid possible problems with the valve action. This can be accomplished without tighter tolerances.

3. The clearance leakage area of note 2. produces a large leak flow rate which is best decreased by use of a smaller nominal diametral clearance of 0.001 in. (or less if your material choices permit it). With 0.001 in. the clearance range would be 0.0007 to 0.0013 in. with use of the same tolerances as at present. If this is done the possiblity of valve malfunctioning would be decreased.

3.1. Leak Rate Past Valve Stem Into Vented Housing Chamber.

The supply pressure gallery in the front plunger sleeve has a leak length of 0.281 in. to the vented chamber. With a supply pressure of 1500 psi, a nil chamber pressure and a diametral clearance of 0.0019 in., a turbulent leak flow calculation leads to a leak flow rate of 0.74 gpm. With the suggested clearance the flow loss could be cut roughly in half.

A-II Exhaust Valve Assembly

1. Exhaust flow areas are adequate, though it would be wise to increase the diameter of the vent hole through the housing if possible. This would decrease the back pressure on the seat end of the valve (see item 3. below).

2. The max. clearance between the valve stem and the sleeve of 0.0018 in. is probably too large. The stem is vented and dirt from the ambient sea will invariably find its way into the clearance. Valve sticking malfunctions will be less frequent with a smaller clearance and the use of anti-lock grooves in the valve stem.

3. Premature Opening During the Drive Stroke. During the drive stroke the cushion shoulder of the plunger displaces water through the poppet spring retainer. This produces a back pressure on the seated end of the poppet. At the same time, the drive chamber pressure decreases due to i) head and inertia losses through the supply poppet and the passage to the chamber and ii) the supply line wave pressure drop caused by the large flow demand of the plunger displacement. The combined drop will be of the order of 1000 psi when no supply line accumulator is used. This could produce a premature opening of the exhaust poppet if the seal on the seated poppet is at the 0.340 in. hole instead of at the 0.498 in. valve seat diameter.

To illustrate, let P_d be the pressure in the poppet sleeve and P_e the back pressure in the spring retainer during the drive stroke. Then, if the valve seals on the 0.34 in. diameter, the force holding the valve closed is

$$F_h = (\pi/4)\{[(.34)^2 - (.311)^2]P_d - (.34)^2 P_e\}.$$

The numerical results presented below demonstrate that the spring force can exceed the hydraulic force and, therefore, that the exhaust valve could open prematurely. If this happened the supply poppet would be affected and the cycle would malfunction.

NUMERICAL RESULTS

P_d (psi)	P_e (psi)	F_h (lb)
1250	0	18.5
	50	14.0
	100	9.5
1000	0	14.8
	50	10.3
	100	5.7
750	0	11.1
	50	6.6
	100	2.0

4. A Short Term Solution. The effect of the back pressure could be diminished if the 0.34 in. hole in the poppet sleeve were chamfered. With a valve seal diameter of 0.375 in. the hydraulic force is 14.8 lb. when P_d = 750 psi and P_e = 100 psi.

5. General Comment. The exhaust poppet valve is an unreliable element of the pre-production prototype drill. The spring force must be low to insure against premature opening during the drive stroke. On the other hand, the spring force should be high to insure that the valve stays open during the return stroke. In addition, it is susceptible to malfunction due to ingestion of ambient dirt particles. On reflection, the use of an exhaust poppet valve was a poor design choice. The operational sensitivity would not be present in a spool valve design because the operating pressures (P_d and P_e) do not influence the valve action.

A-III Supply Valve Assembly.

1. Clearances and Flow Areas. Axial valve travel (valve lift) is sufficient and clearances should cause no problem. The supply hole through the poppet seat has a diameter of 0.380 in. which is smaller than called for in the original design calculations. Thus, the head loss through the poppet seat hole will be higher than originally expected, but this should cause only a very small additional loss in blow energy. The more serious possibility is that any additional head loss will aggravate the situation which could lead to a premature exhaust valve opening during the drive stroke.

2. Valve Opening Condition. The poppet seat hole diameter of only 0.380 in. affects the force balance on the seated poppet valve. The hydraulic force on the seated valve is (under the condition that the drive chamber pressure is zero)

$$F_h = (\pi/4)\{ P_s[(.380)^2 - (.348)^2] - P_k[(.560)^2 - (.380)^2] \}$$
$$= 0.01829\, P_s - 0.1511 P_k .$$

During the return stroke the kicker port pressure, P_k, ultimately decreases so that the hydraulic force becomes positive. When this hydraulic force exceeds the sum of the spring force and seal friction force the valve cracks open. The following numerical results illustrate the situation:

F_h (lb)	P_s (psi)	P_k (psi)	
10	1500	115.4	
	1000	54.8	
	547	0.	Stall
15	1500	82.3	
	1000	21.8	
	820	0.	Stall
20	1500	49.2	
	1094	0	Stall

3. Discussion. The basic design of the supply poppet produces the requirement that the kicker pressure bleed way down before the supply poppet cracks open. This is desired to insure that the piston return stroke is not truncated. Since a large spring force is required to produce a rapid valve closure after piston impact, there could be a problem with stalling as indicated in the numerical results. This is not likely unless an unusually large spring force is in use or the supply pressure at the drill is unusually low. However, it should be considered essential that the supply poppet seal friction be no more than a few ounces. Seal dimensions look all right but the o-ring tolerances may cause problems. Therefore, check the poppet seal friction force before attempting a drill operation. If the static seal friction does not admit essentially free motion of the poppet, change the o-ring or rework the UHMW seal (i.e., decrease the O.D.).

3.1 Kicker Port Pressures. A Design Flaw.

Consider the plunger in its full forward position as shown in the Rock Drill Assembly, Dwg. No. EI-3300-0001-AD. The kicker port galleries are connected to the supply pressure in this position and the supply poppet is closed. This allows the drive chamber to bleed down so the exhaust poppet opens and the return stroke is started. The detail drawings (EI-3300-0020-DD and EI-3300-0017-DD in particular) show that the slot in the plunger stem communicates the kicker port gallery to exhaust pressure when the plunger has moved up 0.93 inches. Lengthy leak flow calculations demonstrate that the kicker pressure remains high enough to keep the supply poppet closed to this point. These calculations were made with the largest clearances around the plunger stem and the supply poppet stem admitted by the specified tolerances.

Further rearward displacement of the plunger communicates the kicker port chamber in the supply poppet assembly to the exhaust (ambient) through the plunger slot and the kicker port gallery in the plunger sleeve. Thus, for example, at a plunger displacement of 0.996 inches the kicker pressure was found to be less than one psi above ambient according to a static leak flow calculation (i.e., fixed plunger position) with a supply pressure of 1500 psi. The calculation makes use of flow continuity and the flow velocity-pressure difference relations

for the exit flow past two orificies in series formed by the plunger slot and the front plunger sleeve, the turbulent leak flow in from the supply gallery in the plunger-sleeve clearance and the laminar leak flow in from supply in the clearance around the poppet stem. Now the potential design flaw is that still further motion of the plunger closes off the kicker port gallery. This occurs at a plunger displacement of 1.062 inches (as shown in the attached Fig. 1) and a static leak flow calculation at this point produces a kicker port pressure of about 300 psi. Still further motion of the plunger leads to even higher kicker port pressures.

These results demonstrate the possibility that the supply poppet will not open during the return stroke or that it could crack open and then close as the kicker port pressure rises to 300 psi and higher. The reason for this possibility is that the kicker port pressure may not have time to decrease to the required low pressure because it communicates to ambient only in a plunger displacement band of 1.062 - 0.93 = 0.132 in. Further, the low pressure is indicated only for a fraction of this span when the plunger displacement is around 0.966 in. In this connection note that the 0.132 in. span is traversed in 0.0011 sec. if the plunger velocity is 10 ft/sec.

3.1.1 The Design Fix.

The design solution to this problem is the extension of the kicker port gallery in the front sleeve as indicated in Fig. 1. This will produce a sustained low kicker port pressure and insure that the supply poppet opens fully and stays open until the drive stroke is well under way.

4. Valve Closing Condition. The plunger in its full forward position does not allow communication of supply pressure to the kicker port gallery. Thus, while supply pressure is established in the kicker port gallery properly during the drive stroke, the supply poppet closing displacement rate cannot be sustained. This will delay (possibly abort) the closing process, so the supply gallery should be extended as shown in Fig. 1 to avoid this possibility. This is important, also, to prevent cavitation in the kicker port gallery of the supply poppet.

A-IV Plunger and Piston Axial Travel Limits.

The numerical results of the section are based on the nominal dimensions in the detail design drawings. Axial tolerances have not been considered. Thus, hardware and assembly dimensional checks should be made, especially since there could be a deleterious stack-up of the axial tolerances in the assembly of the many parts involved (i.e., the housings, sleeves, inserts, plunger, piston and anvil).

1. Plunger. With the plunger in its full forward position the length from the face of the plunger to the face of the plunger sleeve cap (EI-3300-0018-DD) is 1.127 in. The length of the front cushion in the front plunger sleeve is 0.097 in. and the rear cushion length formed by the rear sleeve and cap assembly is 0.063 in. Thus, the total cushion length is 0.16 in. and the free travel of the plunger is only 0.967 in. This is adequate but on the short side as regards the piston blow energy. Operation with 1500 psi at the pump and no supply line accumulator will not produce the design point blow energy.

With the porting dimensions as described above the plunger will always tend to enter the rear cushion. Thus, the porting for the proper functioning of the supply poppet is cut off and the suggested possibility for failure of the supply poppet to open is indicated. In any event, it would be good practice to have the plunger come over top dead center without entering the top cushion. The cushion should be considered a safety feature to prevent plunger impact when there is a malfunction in the cycle operation. (Designing to achieve this feature would require a longer overall drill package, unless the changes suggested in the text are made.)

2. Piston. The axial clearance between the rear face of the piston and the plunger-poppet housing is 1.365 in. when the piston is in its full forward position inside its front cushion. The length of the front cushion is 0.19 in., so a normal impact position 0.125 in. from front cushion entree leaves an axial clearance of $1.365 - 0.19 - 0.125 = 1.05$ in. This indicates that the piston will impact the plunger-poppet housing during normal operation of the drill. Failure of the front end of the plunger-poppet housing will be the result.

A-V Piston Housing - Piston Sleeve Assembly.

The cross holes in the piston housing and the associated piston sleeve serve the purpose of venting the ends of the piston to the ambient. These vents should have a total flow area of the order of the piston cross sectional areas to avoid a head loss from inhibiting the piston motion. The front end is properly vented via four half inch holes directly through the housing. The rear end of the piston, on the other hand, is vented via cross holes in the sleeve insert, an annular gallery and cross holes in the housing. The assembly geometry is shown in Figure 2, and reference to the detail drawings shows that the annular flow area is woefully small. This represents a serious design detail error which was present in the breadboard detail design package as the result of an oversight.

To correct this flaw, additional holes should be added in the housing, but the important change must be to increase the annular flow area. There is room for material removal in both the housing and the sleeve to accomplish this. Ideally the cross-sectional area of the annulus should be half the sum of the four, half inch holes through the sleeve. That is the area should be 0.39 in^2 to prevent unduely large retarding forces during the piston cycling. (Note that the net of the rear piston face area minus the plunger stem area is 0.85 in^2. Thus, a 50 psi pressure rise here would produce a retarding force of 42.5 lb. and this is larger than the design value of the net return stroke force. Proper venting is clearly a most important issue for the proper functioning of the drill.)

A-VI Front Plunger Sleeve - Insert Assembly.

Referring to Drawing No. EI-3300-0020-DD, the interference between the insert and the sleeve can be 0.003 in. With a careless press fit operation the insert could wind up out of round and cause problems with the clearance around the plunger. The latter clearance can be as small as 0.0013 in. according to the design tolerances. Thus, if future designs make use of a tighter plunger - sleeve clearance, the I.D. of the insert should be increased to avoid interference. Alternately, the insert should be installed by heating the sleeve to avoid the press fit operation. In any event the sleeve I.D. should be checked after assembly.

Krasnoff Consulting Associates
107 Kingsway Commons Princeton, N.J. 08540 (609) 497-0895

Mechanical Engineering Specialists

SEAWATER POWERED ROCK DRILL

EVALUATION OF PRESSURE TRACES

Prepared for

EASTPORT INTERNATIONAL
Pacific Marine Systems Division

February 13,1988

SUMMARY

Pressure traces obtained with the seawater powered rock drill have been analyzed and indicate that

i) Excessive leakage in the test hardware in combination with rotation motor flow demand limit the working pressures with the pump system in use.

ii) Lack of a supply line accumulator produces excessive supply line pulsations which will not permit design point power output to be obtained.

iii) Restrictions in the exhaust passages and failure of the exhaust poppet to function properly prevent the achievement of proper cycling.

iv) The failure of the exhaust process to work as desired leads to an excess drive chamber pressure during the return stroke. This upsets the proper functioning of the supply poppet with the result that the return stroke is truncated. That is, the supply poppet opens prematurely.

The traces indicate the necessity for maintaining tight clearances around the plunger and poppets. Also, there may be some leakage when valves are seated and tests are suggested to check this out.

Future tests should be run with the rework suggested in the Design Review Report dated January 28,1988 and with rework of the exhaust passages as indicated in this report. In addition, it is suggested that future tests be run with a supply line of at least 3/4 inch I.D. This will decrease the supply line pulsations and admit higher power output. An alternative would be the use of an in-line accumulator close coupled to the drill.

A final conclusion, generated by the judicious test program results, is that the exhaust valve is the weak link in the drill design concept as suggested in the earlier design review.

EVALUATION OF THE PRESSURE TRACES

ADM 5 Consider a cycle from the point where the kicker pressure, P_k, rises sharply.

1) Following the rise in P_k, the drive chamber pressure, P_d, drops to zero and the supply pressure, P_s, spikes up due to the absence of a supply accumulator. That is, the P_s spike is associated with the sudden drop in supply flow when the supply valve closes. During the time $P_d \approx 0$ the plunger is in its front cushion and the drop in P_s is associated with the flow demand of the piston return shoulder.

2) Following its dwell, P_d jumps when the piston bumps the plunger and the return stroke of the plunger starts. Thereafter P_d holds at a rather high pressure, indicating that the back pressure on the exhaust valve is high, the exhaust valve is not fully open or, possibly, the connection from the drive chamber to and around the exhaust valve is constricted somewhere. (Please check these flow areas.) During the return stroke the supply and drive pressures hold constant, but notice that P_k starts to drift down. This indicates that the plunger has reached the point where the supply - kicker port gallery connection is cut off.

3) Near the low point of P_k the supply valve cracks open. This occurs at a rather high P_k because of the high value of P_d, and causes the return stroke to be truncated. That there is no spike in the drive pressure indicates that the return stroke velocity is very low. The drive pressure just comes up to its relatively constant value during the drive stroke. The drop in supply pressure here is caused by the added flow demand required to pressurize the drive chamber volume and then to sustain the drive stroke. The rise in P_k after its low point is due to the displacement rate of the supply poppet. That P_k follows the supply pressure during the drive stroke indicates that the return stroke is only about half its design value.

4) The end of the drive stroke occurs when the plunger enters its front cushion and decelerates. Here the supply poppet, which has been in a closing mode, approaches its seated position, the drive pressure starts down and the supply pressure spikes to initiate the next cycle.

Discussion

The 0.002 inch coating on the plunger, being severely damaged, could lead to excessive leakage around the plunger. If it is assumed that the clearance around the plunger is 0.002 inches larger than the design clearances, a calculation (using the measured pressures) indicates a cycle averaged leak rate of about 7 gpm, so water must literally be pouring out the exhaust valve exit and the vents above the piston. The very high system flow is the cause of the low operating pressures. During the drive stroke the drive chamber pressure is close to the supply pressure, indicating that the exhaust valve is closed. However, the high chamber pressure during the return stroke indicates that the exhaust valve may not be fully open or that the passages are constricted as suggested above. Another possibility is that the supply poppet seat is not true and allows supply flow into the drive chamber. The poppet seating seal should be checked by fixing the plunger in its full forward position and measuring the drive chamber pressure and the flow out the lower exhaust port.

ADM 9 Basic cycle features are not changed. The extended $P_d = 0$ dwell time may be related to the position of the drill on the test fixture. Thus, it may take longer for the piston to bump the plunger. This would also explain the spike in the drive chamber pressure which is hardly in evidence in the ADM 5 traces. If this is so it means that the return stroke velocity is higher in ADM 9, and this is consistent with the shorter return stroke time. This is one conjecture, but be sure that the impact point is "right on" for all tests.

The high pressure, P_e, in the lower exhaust port indicates a severe flow restriction. Using the measured pressures and the

nominal exhaust valve dimensions, I find that the hydraulic pressure force on the valve during drive stroke is only 12.9 lb. This is pretty low and it is possible that the exhaust valve may not be seated properly. The net force holding it closed is very small so it may seat in a slightly tilted orientation (due to large clearance) and stay put. A check should be made with the plunger fixed in a high position (supply valve open) and with a few hundred psi across the exhaust valve.

ADM 11 The basic cycle sequences appear unchanged and the higher frequency is related to the higher operating pressures. With no rotation motor flow the pump can maintain pressure in spite of the excessive leak rates. However, the lack of a supply line accumulator plays havok with the chamber operating pressures. It would be a good idea to use a supply transducer in all tests so that supply pressure pulsations can be correlated with the cycle chamber pressures. Note the larger drop in P_d during the drive stroke now; this is a reflection that the plunger speed is higher and that the supply poppet is closing during the drive stroke as suggested in the ADM 5 comments.

ADM 12 These traces are very much like those of ADM 9 but appear to represent slightly unsteady cycling. It may be that the force transducer is playing a role here in that it deflects on impact and then recoils. When the time comes there is a precise way in which blow energy can be measured via strain gages mounted on a drill steel. For now the force transducer is adequate to determine changes in blow energy from one build to the next.

ADM 14 With $P_e = 0$ now it is pretty clear that the exhaust poppet does not open. With 200 psi in the drive chamber the pressure force on the valve is about 24 lb. and the spring cannot move the valve. The return stroke occurs with flow displaced around the head of the plunger and, possibly, past an improperly seated exhaust valve. The increased blow energy stems from the increased drive chamber pressure during the drive stroke.

ADM 18 Here it is clear that the exhaust valve never closes. The use of a stiff exhaust poppet spring demonstrates the sensitivity and unreliability of the exhaust poppet design concept. One other feature of these traces is that the drive chamber pressure still rises during the return stroke. This indicates that the flow areas through the exhaust passages are indeed restricted or that the supply poppet is not properly sealed. Thus, the suggested poppet check tests should be made. It is possible that the valves are tilted when seated due to excessive clearance and low seating forces.

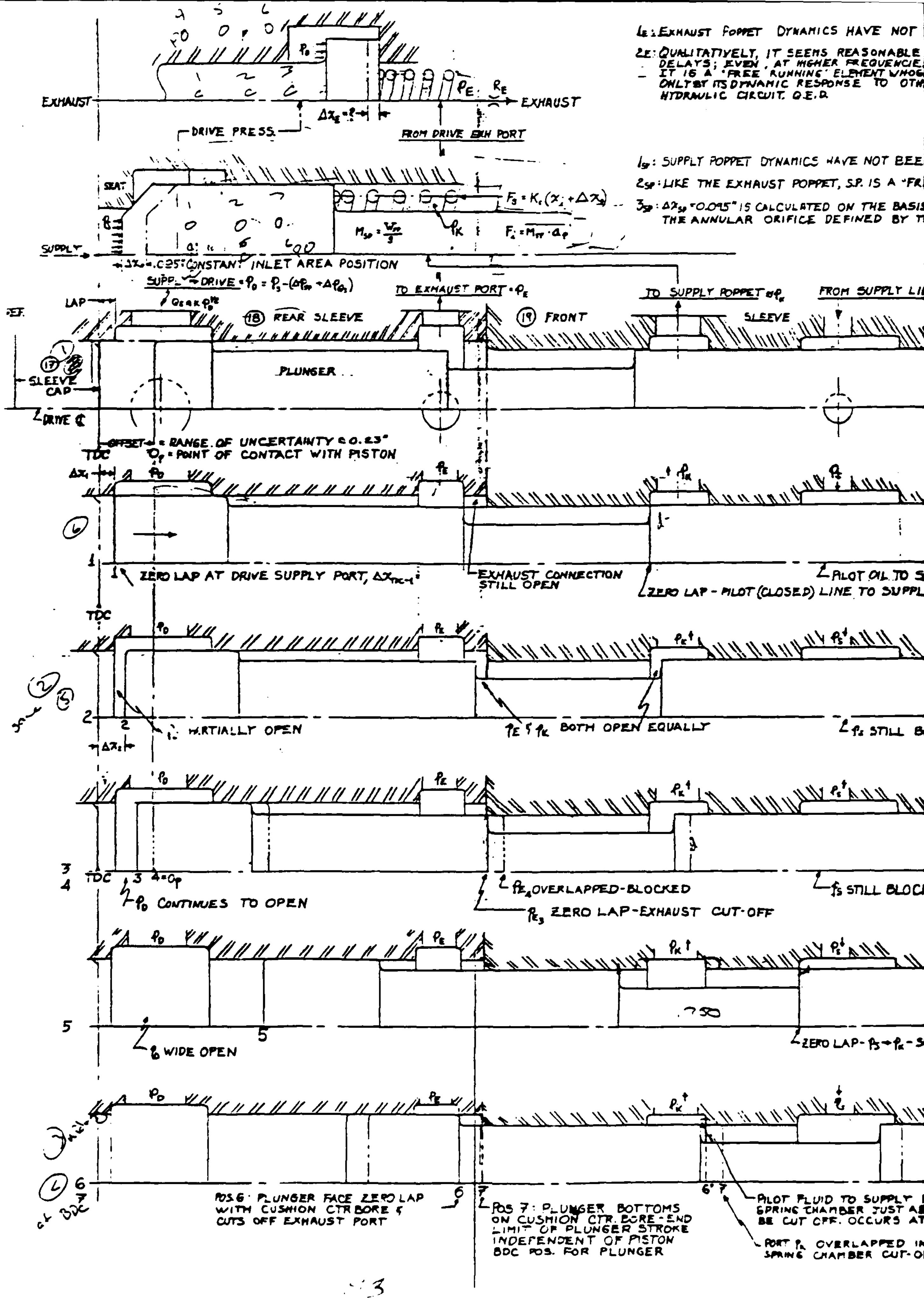
1E: EXHAUST POPPET DYNAMICS HAVE NOT

2E: QUALITATIVELY, IT SEEMS REASONABLE
DELAYS; EVEN, AT HIGHER FREQUENCIE
IT IS A 'FREE RUNNING' ELEMENT WHOS
ONLY BY ITS DYNAMIC RESPONSE TO OTH
HYDRAULIC CIRCUIT. Q.E.D.

1SP: SUPPLY POPPET DYNAMICS HAVE NOT BEE

2SP: LIKE THE EXHAUST POPPET, S.P. IS A "FR

3SP: Δx_{sp} "0.095" IS CALCULATED ON THE BASIS
THE ANNULAR ORIFICE DEFINED BY T



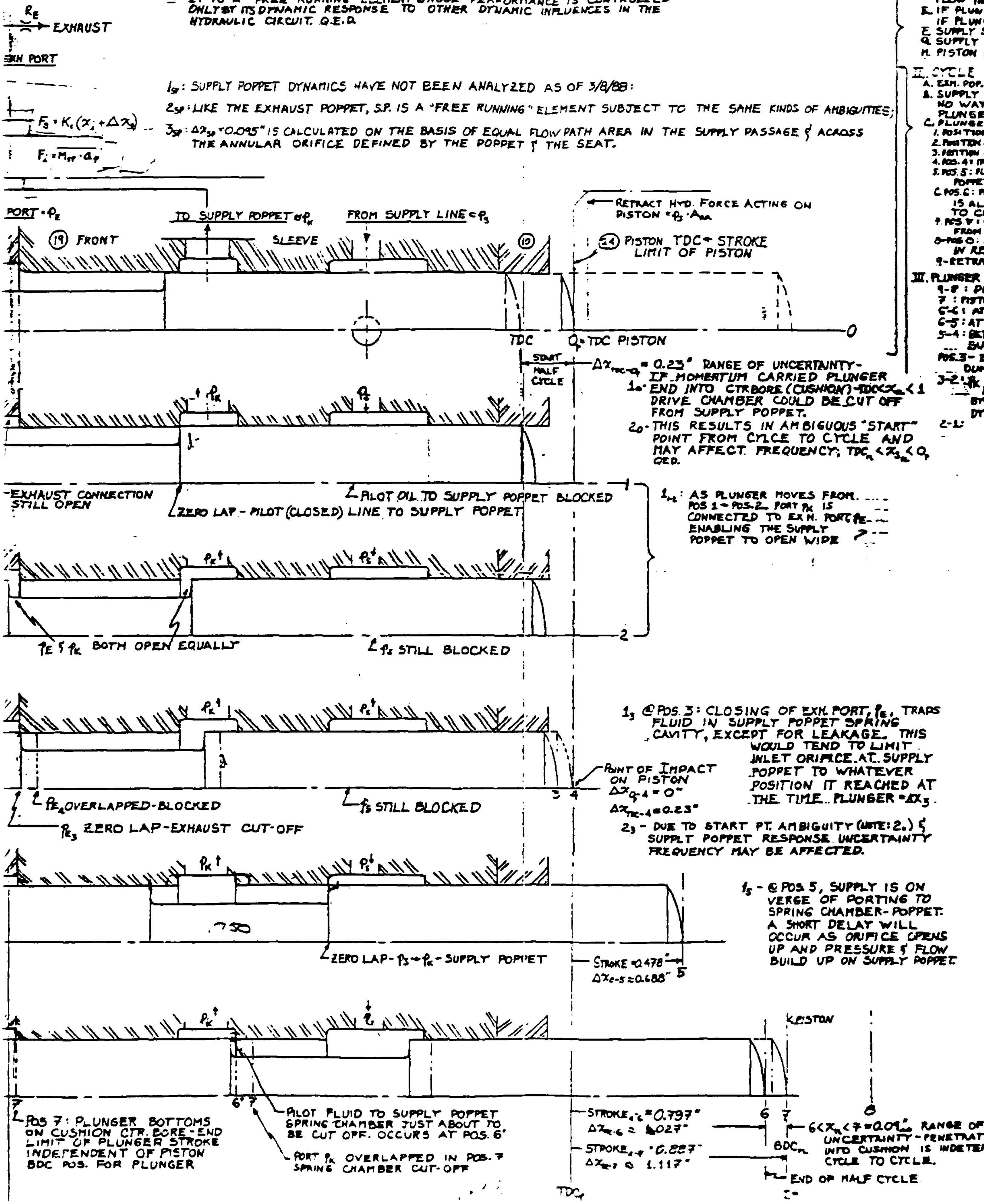
1_E: EXHAUST POPPET DYNAMICS HAVE NOT BEEN ANALYZED – AS OF 3/8/88:

2_E: QUALITATIVELY, IT SEEMS REASONABLE TO ANTICIPATE SOME RESPONSE DELAYS; EVEN, AT HIGHER FREQUENCIES, A TENDENCY TO "HANG" OPEN. IT IS A "FREE RUNNING" ELEMENT WHOSE PERFORMANCE IS CONTROLLED ONLY BY ITS DYNAMIC RESPONSE TO OTHER DYNAMIC INFLUENCES IN THE HYDRAULIC CIRCUIT. Q.E.D.

1_{SP}: SUPPLY POPPET DYNAMICS HAVE NOT BEEN ANALYZED AS OF 3/8/88:

2_{SP}: LIKE THE EXHAUST POPPET, S.P. IS A "FREE RUNNING" ELEMENT SUBJECT TO THE SAME KINDS OF AMBIGUITIES;

3_{SP}: Δx_{SP} = 0.095" IS CALCULATED ON THE BASIS OF EQUAL FLOW PATH AREA IN THE SUPPLY PASSAGE & ACROSS THE ANNULAR ORIFICE DEFINED BY THE POPPET & THE SEAT.



) – AS OF 3/8/88:

SOME RESPONSE TO "HANG" OPEN. :S CONTROLLED JENCES IN THE

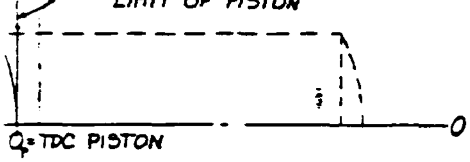
OF 3/8/88:

ENT SUBJECT TO THE SAME KINDS OF AMBIGUITIES;

PATH AREA IN THE SUPPLY PASSAGE & ACROSS : SEAT.

← RETRACT HYD. FORCE ACTING ON PISTON = $p_S \cdot A_{AA}$

2.4 PISTON TDC = STROKE LIMIT OF PISTON



0

Q = TDC PISTON

Δx_{TDC-Q} = 0.23" RANGE OF UNCERTAINTY – IF MOMENTUM CARRIED PLUNGER

1_0 - END INTO CTRBORE (CUSHION) $TDC < x_n < 1$ DRIVE CHAMBER COULD BE CUT OFF FROM SUPPLY POPPET.

2_0 - THIS RESULTS IN AMBIGUOUS "START" POINT FROM CYCLE TO CYCLE AND MAY AFFECT FREQUENCY; $TDC_n < x_{s_n} < 0$, QED.

1

LOCKED

1_{1_2}: AS PLUNGER MOVES FROM POS 1 → POS 2 PORT p_K IS CONNECTED TO EXH. PORT p_E – ENABLING THE SUPPLY POPPET TO OPEN WIDE

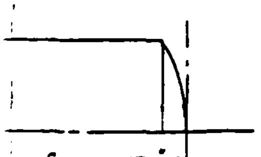
2

1_3 @ POS. 3: CLOSING OF EXH. PORT, p_E, TRAPS FLUID IN SUPPLY POPPET SPRING CAVITY, EXCEPT FOR LEAKAGE. THIS WOULD TEND TO LIMIT INLET ORIFICE AT SUPPLY POPPET TO WHATEVER POSITION IT REACHED AT THE TIME PLUNGER = Δx_3.

POINT OF IMPACT ON PISTON
Δx_{Q-4} = 0"
Δx_{TDC-4} = 0.23"

4

2_3 – DUE TO START PT. AMBIGUITY (NOTE: 2.) & SUPPLY POPPET RESPONSE UNCERTAINTY FREQUENCY MAY BE AFFECTED.

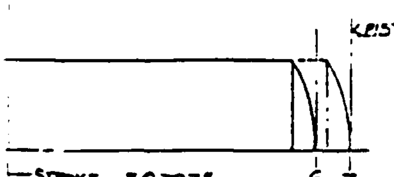


1_5 – @ POS 5, SUPPLY IS ON VERGE OF PORTING TO SPRING CHAMBER-POPPET. A SHORT DELAY WILL OCCUR AS ORIFICE OPENS UP AND PRESSURE & FLOW BUILD UP ON SUPPLY POPPET.

— STROKE = 0.478 →
Δx_{0-5} = 0.688"

5

PISTON



— STROKE$_{0-6}$ = 0.797"
Δx_{TDC-6} = 1.027"

— STROKE$_{0-7}$ = 0.887"
Δx_{TDC-7} ≈ 1.117"

6 7 8

← $6 < x_n < 7$ = 0.09" RANGE OF UNCERTAINTY – PENETRATION INTO CUSHION IS INDETERMINATE, CYCLE TO CYCLE.

BDC_n

END OF HALF CYCLE

I. INITIAL CONDITIONS FOR DRIVE PLUNGER "TIMING" DIAGRAM: POSITION 0.
A. PLUNGER IN RANGE OF UNCERTAINTY BETWEEN TDC & O_P REF. POSITION 0.
B. PLUNGER INSTANTANEOUS VELOCITY $\dot{x}$ = 0 IPS
C. EXHAUST POPPET MUST BE CLOSED @ TDC – OR CLOSING.
D. SUPPLY POPPET IS IN TRANSIENT STATE: $0 < \Delta x_{sp} < 0.095"$ – STARTING TO BUILD PRESSURE & FLOW IN DRIVE CHAMBER. A DELAT IN CYCLE START RESULTS
E. IF PLUNGER IS AT TDC, PORT TO DRIVE CHAMBER IS BLOCKED REF. POS. 0. PLUNGER MAY NOT ST... IF PLUNGER IS AT "0", SUPPLY IS OPEN TO DRIVE CHAMBER.
F. SUPPLY SPRING CHABER, p_K, IS BLOCKED FROM EXHAUST, p_E, BY PLUNGER.
G. SUPPLY LINE, p_S, IS BLOCKED BY PLUNGER
H. PISTON IS AT TDC. MAY OR MAY NOT BE IN CONTACT WITH PLUNGER DEPENDING ON $TDC < x_n < 0$

II. CYCLE STARTS
A. EXH. POP. CLOSED. QED – REF. NOTE: 2E
B. SUPPLY POP. OPENING-ED: REF. NOTE 2sp: IF PLUNGER IS @ TDC, WITHIN THE CUSHION, THERE IS NO WAY TO GET PRESSURE TO THE DRIVE END EXCEPT THROUGH CLEARANCE BETWEEN PLUNGER & CUSHION BORE. QED. MUST OVERRIDE PISTON HYD. RETRACT FORCE. DELAY RESULTS
C. PLUNGER ACCELERATES DOWNWARD DRIVING THE PISTON AHEAD OF IT. POSITIONS 1 – 7.
1. POSITION 1: SUPPLY SPRING CHMBR PORT, p_K, REACHES "ZERO LAP" CONDITION RE. EXHAUST PORT, p_E.
2. POSITION 2: DRIVE CHMBR PORT, p_D, OPENING; p_K CONNECTED TO EXH, p_E – ALLOWING SUPPLY POP. TO OPEN WIDE
3. POSITION 3: DR. CBR. PORT, p_D, OPENING; EXH PORT, p_E, GOES ZERO LAP CUTTING OFF SPRING CBR. PORT, p_K.
4. POS. 4: IF PLGR. POS. WAS $TDC < x_n < O_P$, THE PLUNGER WILL IMPACT ON THE PISTON AT POS. 4
5. POS. 5: PLGR. DRIVES PISTON DOWN; SUPPLY PRESS. p_S PORT "ZERO LAPS" p_K, SPRING CMBR. PORT; POS 5+ SUPPLY POPPET WILL START TO CLOSE
6. POS. 6: PLGR. HAS MOVED TO POINT WHERE IT "ZERO LAPS" EXH. PORT, p_E; SPRING CMBR SUPPLY PORT, p_K, IS ALMOST CUT-OFF; IT WOULD GO TO "ZERO LAP" AT POS. 6"; PLGR. END REACHES ENTRANCE TO CUSHION COUNTERBORE @ POS. 6; SUPPLY POPPET NEAR OR CLOSED. QED
7. POS. 7: PLUNGER STOPS – BOTTOMS ON END OF CUSHION CTR. BORE – BDC; PISTON WILL SEPARATE FROM PLUNGER; IF IT HASN'T YET CONTACTED ANVIL; EXHAUST OPENS BETWEEN POS. 6 & 7, QED;
8-POS 0: SEPARATE POS. OF PISTON – INDETERMINATE, CYCLE TO CYCLE; RESULTS IN A TIME DELAY IN RETRACT OF PLUNGER TO RECYCLE POSITION, $TDC_n < x_n < 0$;
9-RETRACT OF PLUNGER: POS. 7 → POS. 0

III. PLUNGER RETRACT STARTS: REVERSE STEPS II 1-9, ABOVE:
9-8: PISTON HYDRAULICALLY RETRACTED AFTER IMPACTIN ANVIL/TOOL
7: PISTON CONTACTS PLUNGER – PUSHES IT UP; EXHAUST POPPET OPENED BY SPRING. QED.
6-6: AT POS. 6 – SPRG. CHBR. PORT, p_K, IS OPENED TO p_S; THIS HYD. HOLDS THE SUPPLY POPT CLOSED
6-5: AT POS. 5 – P. GR. CUTS OFF p_E FROM p_K, LOCKING SUPPLY POPT. CLOSED HYDRAULICALLY.
5-4: BETWEEN POS 5 & POS. 4 – PISTON CONTINUES TO PUSH PLGR. UP; EXH VALVE REMAINS OPEN; SUPPLY POPPET REMAINS CLOSED; DRIVE CHMB. IS EXHAUSTED THROUGH EXHAUST VALVE
POS. 3 – EXH. PORT p_E, ZERO LAPS WITH PLUNGER LAND; @ POS 3(-), PORT, p_K, STARTS TO DUMP THROUGH p_E, RELEASING SUPPLY POPPET SO IT CAN START TO OPEN. QED
3-2: p_K CONTINUES TO DUMP THRU p_E; SUPPLY CONTS. TO OPEN: EXHAUST VALVE MUST BE OPEN IN ORDER THAT THE PLUNGER NOT LOCK UP – BUT IT MUST CLOSE IN ORDER NOT TO BY-PASS SUPPLY FLUID BEING ROUTED TO THE DRIVE; APPARANTLY SOME CRITICAL DYNAMIC INTERACTIONS OCCUR WHICH WILL HAVE TO BE EVALUATED IN DETAIL. QED
2-1:

MAR 8 1988

PRELIMINARY PRINT

FOR: EASTPORT INT'L – MARINE SYSTEM
VENTURA, CA 93003
by RUSS HENKE ASSOCIATES
WORKSHEET Nº 1: TIMING DIAGRAM
PLUNGER: SEA WATER ROCK DRILL

Russell W. Henke 3/7/88

Appendix E

MP35N AND STELLITE ALLOY 6B MATERIAL SPECIFICATIONS

Alloy DIGEST

MP35N MULTIPHASE

Filing Code: Co-61
Cobalt Alloy

JULY 1970

DATA ON WORLD WIDE METALS AND ALLOYS

Published by
Engineering Alloys Digest, Inc.
Upper Montclair, New Jersey

MP35N* MULTIPHASE*
(Ultra-High Strength Multiphase Alloy)

MP35N MULTIPHASE is a cobalt-nickel-chromium-molybdenum alloy combining ultra-high strength with toughness, ductility, and outstanding corrosion resistance. It is recommended for aerospace fasteners, oceanographic cable, marine hardware, aircraft control cable, torsion bars, pressure vessels, springs, and prosthetic devices and implants.

*Trademarks of Standard Pressed Steel Company, Jenkintown, Pa. U.S. Patent 3356542 and other U.S. and foreign patents pending

Composition:

	Nominal
Cobalt	35
Nickel	35
Chromium	20
Molybdenum	10

Physical Constants:

Density, lb/cu in.		0.304
Specific gravity		8.43
Thermal coef. expansion/°F		
	70 to 200°F	7.1×10^{-6}
	70 to 600°F	8.2×10^{-6}
	70 to 800°F	8.3×10^{-6}
Melting range, °F		2400–2625
Modulus of elasticity, psi		
	70°F	33.6×10^6
	400°F	31.9×10^6
	800°F	29.6×10^6
Electrical resistivity, microhm-cm (±2%)		105
Magnetic properties		Non-magnetic

PROPERTIES

Table 1 – TYPICAL MECHANICAL PROPERTIES

	Annealed	Work Strengthened and Aged		
Tensile strength, psi	146000	200000	260000	300000
Yield Strength (0.2% Offset), psi	61000	160000	225000	290000
Elongation (4 D), %	70	18	11	9
Reduction of area, %	70	60	55	43
Rockwell C hardness	8	45	49	54
Charpy impact (V-notch), ft-lb	200+	95	23	17
Endurance limit*, psi	–	–	130000	–

*Tension – Tension fatigue. Stress ratio = 0.1 smooth bar.

Table 2 – EFFECT OF COLD REDUCTION AND AGING ON HARDNESS

Cold Reduction %	Rockwell C Hardness: As Cold Reduced	Cold Reduced and Aged 4 Hours at 1000°F	1100°
0 (Annealed)	9	9	9
10	24	28	28
20	32	37	37
30	37	42	42
40	41	46	46
50	45	50	50
60	48	53	53
70	51	55	55

Table 3 – EFFECT OF AGING TEMPERATURE ON ROOM-TEMPERATURE TENSILE PROPERTIES
(Starting material cold reduced 48.5% and aged 4 hours)

Aging Temperature °F	Tensile Strength psi	Yield Strength (0.2% Offset) psi	Elongation %	Reduction of Area %
As cold drawn	248000	198000	12	60
800	278000	250000	10	55
900	280000	260000	9	50
1000	290000	275000	10	50
1100	280000	270000	11	50
1200	270000	260000	12	52

Table 4 – EFFECT OF COLD REDUCTION AND AGING ON ROOM-TEMPERATURE TENSILE PROPERTIES

Cold Reduction %	Tensile Strength psi	Yield Strength (0.2% Offset) psi	Elongation %	Reduction of Area %
As Cold Reduced				
15	155000	118000	40	70
25	170000	150000	28	70
35	215000	170000	20	63
45	240000	180000	15	58
55	265000	195000	13	57
65	285000	235000	10	54
Cold Reduced and Aged at 1000°F for 4 Hours				
15	157000	125000	38	70
25	186000	175000	27	66
35	235000	220000	13	60
45	285000	270000	12	54
55	320000	310000	10	40
65	360000	350000	4	[illegible]

Table 5 – RECOMMENDED CONDITIONS FOR MACHINING MP35N ALLOY WORK STRENGTHENED TO A TENSILE STRENGTH OF 260000 PSI

Operation	Tool Mat'l	Tool Geometry	Type of Tool	Depth of Cut, in.	Width of Cut in.	Feed	Cutting Speed ft/min.	Tool Life	Wear land in.	Cutting Flu[id]
Turning	M42 HSS	BR: 0° ECEA: 15° SR: 10° Rel: 5° SCEA: 15° NR: .030"	5/8" square tool bit	0.050	–	0.010 in./rev.	30	60 min. plus	0.020	Soluble Oil (1:20)
Peripheral End Milling	M2 HSS	Helix Angle: 30" RR: 10° CA: 45° x .060" Per. Cl: 7°	1" dia. 4 flute end mill	0.125	0.500	0.002 in./tooth	75	200" work travel	0.005	Sulfurized Oil
Drilling	T15 HSS	Point Angle: 118° Helix Angle: 29° Cl: 7° Point: Crankshaft	1/4" dia. 2 flute drill screw machine length	1/2" through	–	0.005 in./rev.	25	250 holes plus	0.012	Chlorinated Oil
Reaming	M2 HSS	Straight Flute Chamfer Angle: 45° Relief: 7°	Letter I dia. (.272") 6 flute HSS reamer	1/2" through	–	0.009 in./rev.	60	195 holes	0.006	Chlorinated Oil
Tapping	M1 HSS	2 Flute Plug Spiral Point 75% Thread	5/16-24 NF tap	1/2" through	–	–	5	235 holes	Tap breakage or undersize thread	Chlorinated Oil

Heat Treatment:

Annealing: MP35N MULTIPHASE alloy is annealed by heating to 1925-2000°F for 1-2 hours and air cooling to room temperature to produce a maximum hardness of about Rockwell C20.

Aging: Heating in the 800 to 1200°F range after cold working produces further increases in hardness and strength. For most applications, aging at 1000 to 1100°F for 4 hours after work strengthening provides the optimum combination of strength and ductility. Aging is effective only when it follows work strengthening. Aging annealed material will produce no increase in strength.

Machinability:

Recommended machining practice is given in Table 5. Surface grinding can be done with an alumina wheel 10 x 1 x 3 inches (Grade 32A46J8VBE), down feed 0.002 in./pass, cross feed 0.050 in./pass, table speed 40 ft/min., wheel speed 6000 ft/min., G Ratio 70, using sulfurized oil as grinding fluid.

Workability:

Cold or warm working is used to strengthen the MP35N alloy, by rolling, swaging, cold extrusion, or drawing. Strength and hardness increase nearly linearly with per cent cold work, although the material retains excellent ductility even with large amounts of cold work.

Weldability:

MP35N has been tested under various welding methods and has shown excellent results. Data reported to date suggest that MP35N has weldability characteristics similar to Type 304 Stainless Steel. The following welding parameters were developed with the TIG process on 1/4" thick plate specimens in a butt welding configuration:

Welding Speed	5 1/2 ipm
Current	100 - 160 amps
Voltage	10 volts
MP35N Filler Wire Feed	16 – 20 ipm
Argon Gas Flow Rate	25 cfh

Corrosion Resistance:

MP35N MULTIPHASE alloy is resistant to most mineral acids, sea water and salt spray environments. It has shown immunity to stress corrosion in boiling 42% magnesium chloride and modified 10% sodium chloride tests. The alloy also is immune to crevice and stress corrosion in sea water or synthetic seawater. Coupons of the alloy were immersed in sea water for more than four years and remained bright and free from corrosion products. This alloy appears to be completely resistant to sea water corrosion regardless of process condition or strength level. It also appears to be practically immune to stress-corrosion cracking.

Specification Equivalents:

AMI 13 (7-1-69)
SPS-M-573

General Characteristics:

MP35N MULTIPHASE is a cobalt-nickel-chromium-molybdenum alloy that has a unique combination of properties – ultra high strength, toughness, ductility and outstanding corrosion resistance. The alloy is hardened by work strengthening and aging to strength levels of 260000 to 300000 psi.

MP35N MULTIPHASE alloy has a face centered cubic matrix of cobalt and nickel in which the chromium and molybdenum are soluble at elevated temperatures. The face centered cubic structute persists upon cooling to room temperature and below. Working the alloy at temperatures below the equilibrium transformation temperature (approximately 850°F) causes local shear transformation to form very small platelets of the hexagonal close packed structure. The transformation does not appear to have an M_s temperature at which it occurs on cooling, as does martensite in steel. It does occur, however, as a strain induced transformation – the amount of transformed product being a function of the amount of strain deformation. Work strengthening can be accomplished by extruding, rolling, swaging, drawing or a combination of these manufacturing processes. The transformation occurs readily with work at room temperature, but will also occur at elevated temperatures to the upper limit of the transformation zone. The hexagonal close packed platelets that are formed are stable in the face centered cubic matrix and the resultant structures exhibit the unique combination of excellent mechanical properties and corrosion resistance. Transformation strengthened material is usually aged to obtain even higher strength levels through precipitation strengthening.

The new alloy performs well at cryogenic temperatures and is recommended for service to 700°F.

Forms Available:

Bar, rod, wire and tubing.

Applications:

Fasteners, cables, marine hardware, torsion bars, and springs.

Manufacturer:

Latrobe Steel Company
Latrobe, Pennsylvania 15650

Alloy DIGEST — DATA ON WORLD WIDE METALS AND ALLOYS

Filing Code: Co-23
Cobalt Alloy

SEPTEMBER 1960

Published by
Engineering Alloys Digest, Inc.
Upper Montclair, New Jersey

Haynes Stellite Alloy No. 6B

HAYNES STELLITE ALLOY No. 6B
(High Temperature Alloy)

HAYNES STELLITE ALLOY No. 6B is a cobalt-base alloy recommended for handling extreme conditions of wear, abrasion, and heat.

(The term Haynes Stellite is a registered trademark of Union Carbide Corp.)

Composition:

	Nominal
Carbon	1.1
Manganese	2.0 max.
Silicon	2.0 max.
Iron	3.0 max.
Molybdenum	1.5 max.
Nickel	3.0 max.
Chromium	30.0
Tungsten	4.5
Cobalt	Remainder

Physical Constants: (at 70 deg. F.)

Specific gravity	8.38
Density, lb/cu. in.	0.303
Specific heat, BTU/lb/°F	0.101
Thermal conductivity, BTU/ft²/in/hr/°F	102.7
Thermal coef. expansion, in/in/°F x 10^{-6}	
32- 212 °F	7.7
32- 932 °F	8.3
32-1832 °F	9.7
Electrical conductivity, % IACS	1.9
Electrical resistivity, microhms-cm	91.0
Modulus of elasticity, psi x 10^6	30.4
Modulus of rupture, psi x 10^3	388
Melting range, °F	2310-2470

PROPERTIES

Table 1 — TYPICAL MECHANICAL PROPERTIES — HOT ROLLED

	1/8" Sheet	1/2" Plate
Tensile strength, psi	165000	148000
Yield strength, psi (0.2%)	110000	88000
Elongation, % in 2"	5	7
Reduction of area, %	—	9
Compressive strength, psi	—	347600
Transverse strength, lbs. 4" span, 1.2" sq. bars	7050	—
Rockwell hardness	C43	C38
Izod impact, ft. lbs. (unnotched)		
Long.	—	62
Trans.	—	57
Charpy impact, ft. lbs.		
Unnotched Long.	—	72
Unnotched Trans.	—	65
Notched Long.	—	6

(Sheet—Hot rolled 0.125 inch thick.)
(Plate—Mill-annealed plate at 2250°F, rapid air-cooled.)

Table 2 — ELEVATED TEMPERATURE PROPERTIES

(Specimens cut from mill-annealed 1/2" plate and machined to round test bars with 1/4" dia. section. Longitudinal axis of specimens was parallel to rolling direction.)

Test Temperature °F	Tensile Strength psi	Yield Strength psi (0.2%)	Elongation % in 1"	Reduction of Area %
1000	133000	58500	9	15.0
1250	115000	60000	9	15.5
1500	93000	49000	13	26.0

Table 3 — CHARPY IMPACT DATA

Test Temperature °F	Rolling Direction Longitudinal	Charpy Impact ft.lbs.
1000	Unnotched	81
	Notched	15
1250	Unnotched	116
	Notched	15
1500	Unnotched	126
	Notched	15

Heat Treatment:

This alloy is normally supplied in the as-hot rolled condition and can be given a solutioning-type heat treatment by heating at 2250 deg. F. followed by a rapid air-cool. A solution treatment may be desirable when maximum corrosion resistance with abrasion resistance is sought. The cast form of this alloy is less sensitive to this type of treatment than is the wrought form. It cannot be hardened or strengthened by any thermal treatment. A heat treatment of 4 hours at 1650 deg. F., followed by a furnace-cool is recommended for maximum machinability. In order to stress relieve the alloy, charge into a cold furnace, heat slowly to 1650 deg. F., for for at least 2 hours and then allow to cool in the furnace. It may be necessary to adjust the time to suit the size of part being heat-treated.

Machinability:

Can be satisfactorily machined with carbide-tipped tools and water soluble oil cutting fluid. Cutting edges for turning and facing tools should have 5 deg. primary clearance, 10 deg. secondary clearance and lead angle of 45 deg. Use Carboloy 905 or equivalent. For rough turning use cutting speeds of 30-40 sfpm with 0.008-0.015 ipr feed and 0.040-0.050" depth of cut. For finishing use cutting speeds of 30-45 sfpm with 0.005-0.008 ipr feed and 0.010-0.025" depth of cut. For drilling use carbide tipped drills or masonry type with 1 part soluble oil and 1 part kerosene. For best results keep drill web as thin as possible and operate at 20-35 sfpm cutting speeds with 0.002-0.005 ipr feed.

Weldability:

Readily welded by oxy-acetylene, metallic-arc, and Heli-arc methods. When welding, preheat to a cherry red heat (about 1100 deg. F.) in order to avoid cracking. It can be joined to other materials by brazing with silver solder, or copper brazing. Slow post cooling after welding should be used to avoid cracking.

Workability:

This alloy is somewhat limited in its ease of fabrication because of its high strength. Special forming operations can be performed at temperatures above the bright red range (minimum 1800 deg. F.). For wall thicknesses greater than ¼", it is generally more practical to use cast forms.

Corrosion Resistance:

Corrosive Media	10% Conc. at Room Temp.	10% Conc. at Boiling Point	Concentrated at Boiling Point	Saturated Vapor at Room Temp.
Wet Chlorine				U
Acetic Acid		U	U	
Nitric Acid		E	F	
Sulfuric Acid	U	F	*P	
Phosphoric Acid (30%)			G	
Ferric Chloride	U	G		
Ferric Sulfate	U	U		
Sodium Hydroxide (30%)			G	

U — Unaffected
E — Excellent — less than 0.001" penetration per year
G — Good — less than 0.010" penetration per year
F — Fair — less than 0.100" penetration per year
P — Poor — greater than 0.100" penetration per year.
*77% concentration at boiling point

General Characteristics:

This cobalt-base alloy has high heat, abrasion, and wear resistance. It has low coefficient of friction and is non-galling. It retains high hardness at red heat and recovers full hardness after exposure to temperatures as high as 2000 deg. F. Its resistance to oxidation and corrosion is excellent. Alloy No. 6B is tough and strong. It also has good impact strength, and is resistant to heat checking at elevated temperatures and to thermal shock.

Forms Available:

Sheet, plate, and fabricated items.

Applications:

Erosion shields for turbine blades, valve parts, valve seat inserts, scarfing machine blow pipe shoes, hot work punches, half and full sleeves, half and full bushings, scraper bars, metal cutting saws, cylinder liners, wear strips, surgical blades, and mirrors

Manufacturer:

Haynes Stellite Company
Division of Union Carbide Corporation
Kokomo, Indiana

DISTRIBUTION LIST

CBC / Tech Lib, Davisville, RI
CBC / Tech Lib, Gulfport, MS
CNA / Tech Lib, Alexandria, VA
COMDT COGUARD / Lib, Washington, DC
CORNELL UNIV / Civil & Environ Engrg, Ithaca, NY
DNA / Tech Svcs Lib, Mercury, NV
DOE / Knolls Atomic Power Lab, Lib, Schenectady, NY
DTIC / Alexandria, VA
EASTPORT INTL, INC / S. Anderson, Mgr, Ventura, CA
FLORIDA ATLANTIC UNIV / Ocean Engrg Dept (McAllister), Boca Raton, FL
GIDEP / OIC, Corona, CA
NAVCOASTSYSCEN / Code 2360, Panama City, FL
NAVCOASTSYSCEN / Code 715 (J. Mittleman), Panama City, FL
NAVCOMMSTA / Lib, FPO San Francisco,
NAVEODTECHCEN / Tech Lib, Indian Head, MD
NAVFACENGCOM / Codes 03, 03T (Essoglou), 04A, 07, 07A (Herrmann), 09M124 (Lib), Alexandria, VA
NAVFACENGCOM CHESDIV / FPO-1PL, Washington, DC
NAVFACENGCOM LANTDIV / Lib, Norfolk, VA
NAVFACENGCOM SOUTHDIV / Lib, Charleston, SC
NAVOCEANSYSCEN / Code 94, San Diego, CA
NAVOCEANSYSCEN / DET, Tech Lib, Kailua, HI
NAVPGSCOL / Codes 68 (C.S. Wu), 68WY (Wyland), 69 (T. Sarpkaya), Monterey, CA
NAVPGSCOL / E. Thornton, Monterey, CA
NTIS / Lehmann, Springfield, VA
OFFICE OF SEC OF DEFENSE / DDR&E, Washington, DC
OKLAHOMA STATE UNIV / Ext Dist Offc. Tech Transfer Cen, Ada, OK
PENNSYLVANIA STATE UNIV / Rsch Lab, State College, PA
PWC / Code 123C, San Diego, CA
PWC / Code 134 Lib, Pearl Harbor, HI
PWC / Lib, Norfolk, VA
PWC / Lib, Pensacola, FL
SCRIPPS INST OF OCEANOGRAPHY / Lib, LaJolla, CA
TEXAS A&M UNIV / CE Dept (Machemehl), College Station, TX
UCT ONE / CO, Norfolk, VA
UCT TWO / CO, Norfolk, VA
UNITED KINGDOM / Inst of Oceanographic Sci, Wormely, Godalming, Surrey
UNIV OF ALASKA / Biomed & Marine Sci Lib, Fairbanks, AK
UNIV OF CALIFORNIA / CE Dept (Gerwick), Berkeley, CA
UNIV OF HARTFORD / CE Dept (Keshawarz), West Hartford, CT
UNIV OF ILLINOIS / Arch School (Kim), Champaign, IL
UNIV OF MISSOURI / Military Sci Dept, Rolla, MO
UNIV OF NEW MEXICO / NMERI (Falk), Albuquerque, NM
UNIV OF TEXAS / Fusion Studies Inst (Kotschenreuther), Austin, TX
USNA / Rsch Dir, Annapolis, MD

INSTRUCTIONS

The Naval Civil Engineering Laboratory has revised its primary distribution lists. To help us verify our records and update our data base, please do the following:

- Add - circle number on list
- Remove my name from all your lists - check box on list.
- Change my address - line out incorrect line and write in correction **(DO NOT REMOVE LABEL).**
- Number of copies should be entered after the title of the subject categories you select.
- Are we sending you the correct type of document? If not, circle the type(s) of document(s) you want to receive listed on the back of this card.

Fold on line, staple, and drop in mail.

DEPARTMENT OF THE NAVY

Naval Civil Engineering Laboratory
Port Hueneme, CA 93043-5003

Official Business
Penalty for Private Use, $300

NO POSTAGE NECESSARY IF MAILED IN THE UNITED STATES

BUSINESS REPLY CARD
FIRST CLASS PERMIT NO. 12503 WASH D.C.

POSTAGE WILL BE PAID BY ADDRESSEE

Commanding Officer
Code L34
Naval Civil Engineering Laboratory
Port Hueneme, CA 93043-5003

DISTRIBUTION QUESTIONNAIRE

The Naval Civil Engineering Laboratory is revising its primary distribution lists.

SUBJECT CATEGORIES

1 SHORE FACILITIES
1A Construction methods and materials (including corrosion control, coatings)
1B Waterfront structures (maintenance/deterioration control)
1C Utilities (including power conditioning)
1D Explosives safety
1E Aviation Engineering Test Facilities
1F Fire prevention and control
1G Antenna technology
1H Structural analysis and design (including numerical and computer techniques)
1J Protective construction (including hardened shelters, shock and vibration studies)
1K Soil/rock mechanics
1L Airfields and pavements
1M Physical security
2 ADVANCED BASE AND AMPHIBIOUS FACILITIES
2A Base facilities (including shelters, power generation, water supplies)
2B Expedient roads/airfields/bridges
2C Over-the-beach operations (including breakwaters, wave forces)
2D POL storage, transfer, and distribution
2E Polar engineering
3 ENERGY/POWER GENERATION
3A Thermal conservation (thermal engineering of buildings, HVAC systems, energy loss measurement, power generation)
3B Controls and electrical conservation (electrical systems, energy monitoring and control systems)
3C Fuel flexibility (liquid fuels, coal utilization, energy from solid waste)
3D Alternate energy source (geothermal power, photovoltaic power systems, solar systems, wind systems, energy storage systems)
3E Site data and systems integration (energy resource data, integrating energy systems)
3F EMCS design
4 ENVIRONMENTAL PROTECTION
4A Solid waste management
4B Hazardous/toxic materials management
4C Waterwaste management and sanitary engineering
4D Oil pollution removal and recovery
4E Air pollution
4F Noise abatement
5 OCEAN ENGINEERING
5A Seafloor soils and foundations
5B Seafloor construction systems and operations (including diver and manipulator tools)
5C Undersea structures and materials
5D Anchors and moorings
5E Undersea power systems, electromechanical cables, and connectors
5F Pressure vessel facilities
5G Physical environment (including site surveying)
5H Ocean-based concrete structures
5J Hyperbaric chambers
5K Undersea cable dynamics
ARMY FEAP
BDG Shore Facilities
NRG Energy
ENV Environmental/Natural Responses
MGT Management
PRR Pavements/Railroads

TYPES OF DOCUMENTS

D = Techdata Sheets; **R** = Technical Reports and Technical Notes; **G** = NCEL Guides and Abstracts; **I** = Index to TDS; **U** = User Guides; ☐ None - remove my name

NCEL DOCUMENT EVALUATION

You are number one with us; how do we rate with you?

We at NCEL want to provide you our customer the best possible reports but we need your help. Therefore, I ask you to please take the time from your busy schedule to fill out this questionnaire. Your response will assist us in providing the best reports possible for our users. I wish to thank you in advance for your assistance. I assure you that the information you provide will help us to be more responsive to your future needs.

RNStorer

R. N. STORER, Ph.D, P.E.
Technical Director

DOCUMENT NO. ________ TITLE OF DOCUMENT: ________________________

Date: ________ Respondent Organization : ________________________

Name: ________________ Activity Code: ________________

Phone: ________________ Grade/Rank: ________________

Category (*please check*):

Sponsor ____ User ____ Proponent ____ Other (*Specify*) ________________ ____

Please answer on your behalf only; not on your organization's. Please check (*use an X*) only the block that most closely describes your attitude or feeling toward that statement:

SA Strongly Agree   A Agree   O Neutral   D Disagree   SD Strongly Disagree

---

	SA	A	N	D	SD
1. The technical quality of the report is comparable to most of my other sources of technical information.	( )	( )	( )	( )	( )
2. The report will make significant improvements in the cost and or performance of my operation.	( )	( )	( )	( )	( )
3. The report acknowledges related work accomplished by others.	( )	( )	( )	( )	( )
4. The report is well formatted.	( )	( )	( )	( )	( )
5. The report is clearly written.	( )	( )	( )	( )	( )
6. The conclusions and recommendations are clear and directly supported by the contents of the report.	( )	( )	( )	( )	( )
7. The graphics, tables, and photographs are well done.	( )	( )	( )	( )	( )

Do you wish to continue getting NCEL reports?	☐ YES	☐ NO

Please add any comments (e.g., in what ways can we improve the quality of our reports?) on the back of this form.

Comments:

Please fold on line and staple

DEPARTMENT OF THE NAVY

Naval Civil Engineering Laboratory
Port Hueneme, CA 93043-5003

Official Business
Penalty for Private Use $300

Code L03B
NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME, CA 93043-5003